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THE CORROSION OF STEEL AND VARIOUS ALLOYS BY HIGH TEMPERATURE STEAM.

BY H. L. SOLBERG,[1] G. A. HAWKINS,[2] AND A. A. POTTER.[3]

Central-station installations now under construction or in operation in the high-pressure range, about 1400 psi steam pressure, are designed to operate on steam between 900 F. and 950 F. Better efficiency would be obtained by using even higher steam temperatures. Since the inner surface of a superheater tube, subject to heat transfer, is at a higher temperature than the steam in the tube, it is apparent that superheater tubes may be operated at temperatures which approach those used in the commercial production of hydrogen from steam. A knowledge of the extent

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to which steam reacts with alloy steels at these temperatures is of considerable value in determining the best material for service under high-temperature conditions. The hot end of the high-temperature superheater is now made of alloy steel, but few data are available as to the effect of steam on the various alloy steels that have been proposed for this service.

The investigation which has been carried on at Purdue University since 1934 has had for its objectives the determination of the effect of steam pressure, types of surface shapes and finishes, time of exposure to high temperature steam, effect of temperature between 1000 F. and 1300 F., and the influence of temperature fluctuations on the corrosion resistance and spalling of alloy steel samples.

Considerable time was devoted to experimental work in order to develop suitable equipment and testing technique. In general, the method used for measuring the corrosion products consists of surface sandblasting a sample of steel to insure uniform surface finish, accurately weighing the sandblasted sample, placing the sample in a reaction chamber which is maintained at the same temperature as the steam, removing the test specimen and stripping the scale electrolytically using a solution of acid and an inhibitor, and finally reweighing the stripped specimen. The difference in initial and final weights represents the loss in weight of metal, which is an index of the amount of corrosion.

Effect of Temperature Upon the Corrosion of Low-Carbon Steel Tubes by Steam.

A series of tests were conducted[1] at a constant steam pressure of 1200 psi gauge with steam and metal surface temperatures from 800 F. to 1200 F. Table I shows the results for this test. The high temperature test was terminated at the end of 23 hours because of the failure of a stop valve which had been subjected to rapid fluctuations in temperature when cutting in and shutting off steam from the test sections. These data are plotted in Figure 1, and may be represented between 800 F. and 1100 F. by the equation

$$W = 5.56\ (10^{-7})\ e^{0.011t} \qquad (2)$$

where W represents the scale formed in grams per square inch of internal tube surface and t the surface temperature in degrees F.

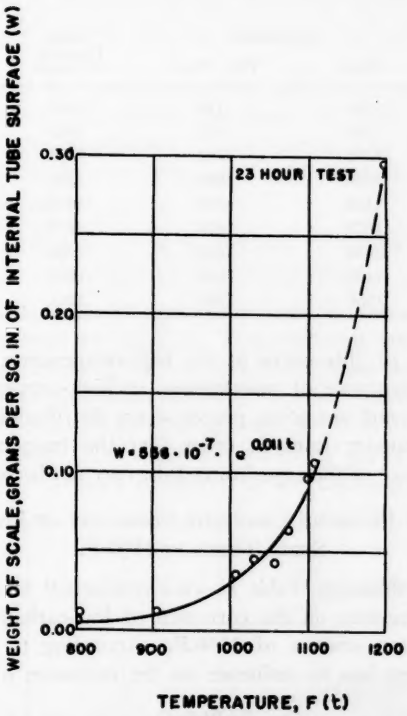

FIGURE 1. EFFECT OF TEMPERATURE ON THE WEIGHT OF SCALE FORMED (THE CURVE IS BASED ON 36-HOUR TESTS OF LOW-CARBON-STEEL TUBES WHICH WERE SUBJECTED TO A STEAM PRESSURE OF 1200 PSI.)

While the general location of the curve above 1100 F. is questionable because of the shortage of the test at 1200 F., it is probably steeper than indicated in Figure 1; and the marked influence of temperature upon the corrosion of low-carbon steel by steam is clearly indicated.

TABLE I.

EFFECT OF TEMPERATURE UPON THE CORROSION OF LOW-CARBON STEEL TUBES BY STEAM.

Test Number	Temperature, F.		Steam Pressure, psi gauge	Scale formed in 36 hours, grams per sq. in. of internal tube surface
	Steam	Tube Metal		
9	801	794	1195	0.0101
7	904	894	1200	0.0125
16	1005	1001	1200	0.0370
15	1026	1023	1200	0.0452
11	1055	1045	1199	0.0432
14	1073	1072	1200	0.0642
18	1099	1097	1200	0.0973
19	1105	1099	1200	0.1065
17	1200	1195	1201	0.2970*

* 23-hour test.

The slope of this curve at the high-temperature end emphasizes the importance of maintaining, in high-temperature superheater design and operation, proper steam distribution and reasonable heat transfer rates in order that the temperature of the internal surface of the superheater tube may not become excessive.

EFFECT OF PRESSURE UPON THE CORROSION OF LOW-CARBON STEEL TUBES AT 1100 F.

The tests shown in Table II were conducted[1] to study the influence of pressure on the corrosion of low-carbon steel tubing at a steady temperature of 1100 F. According to these results steam pressure has no influence on the corrosion of low-carbon

TABLE II.

EFFECT OF STEAM PRESSURE UPON THE CORROSION OF LOW-CARBON STEEL BY STEAM AT 1100 F.

Test Number	Steam Pressure, psi gauge	Temperature, F.		Scale formed in 36 hours, grams per sq. in. of internal tube surface
		Steam	Metal	
10	400	1100	1097	0.0962
21	397	1098	1095	0.1046
18	1200	1099	1097	0.0973
19	1200	1105	1099	0.1065

steel between 400 and 1200 psi gauge. In view of the absence of pressure effects and the expense of operating high-pressure steam generating equipment, it was decided to revise the test apparatus and to superheat steam from the regular laboratory steam header where the normal operating pressure is 200 psi gauge.

The following test results constitute part of the paper entitled "The Corrosion Of Unstressed Specimens Of Steel And Various Alloys By High Temperature Steam" which was presented by the authors at the October, 1941, meeting of the American Society of Mechanical Engineers.

Test on Unstressed Bar Specimens to Determine the Influence of Length of Test.

The effect of time on corrosion at 1100 F. for a number of different alloy bars was studied. Four different tests were conducted on the various alloys; the lengths of these tests were 200 hours, 500 hours, 1000 hours, and 2000 hours. Specimens of each alloy steel and low carbon steel as listed in Table III were used in each of the tests. The smoothed results are shown in Figure 2. In general the corrosion decreases as the chromium content of

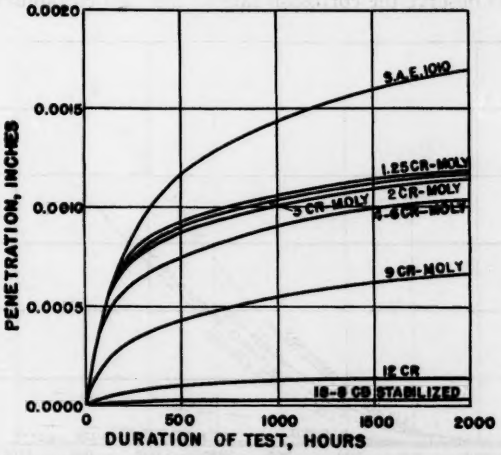

FIGURE 2. CORROSION OF STEEL BARS IN CONTACT WITH STEAM AT 1100 F FOR 200, 500, 1000, AND 2000 HOURS

the steel increases. The high chromium steels are practically corrosion resistant. The various 4-6 Cr steels fall in one general class, and it is not possible from this test to segregate the steels within this group. The corrosion is very rapid during the first 100 hours after which the layer of corrosion products retards the oxidation rate under steady temperature conditions. During this series of tests, from 65 to more than 80 per cent of the corrosion occurring in 2000 hours took place during the first 500 hours.

Effect of Steam Temperature on the Corrosion of Bar Specimens.

A series of tests was completed to determine the effect of steam temperature for a constant time on the corrosion resistance of the same steels as listed in Table III, together with samples of carbon-moly steel. The specimens were treated in the usual manner and subjected to steam at various temperatures. The smoothed results are shown as solid lines in Figure 3. Except for the high chromium steels corrosion increases very rapidly at temperatures in excess of about 1100 F. The high chromium steels showed very little corrosion even at 1200 F.

A special test was conducted at very high steam temperatures in order to observe the corrosion rate of the high chromium steels.

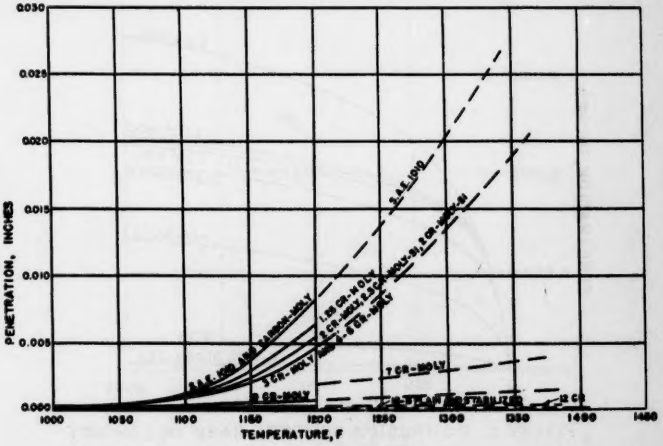

FIGURE 3. CORROSION OF STEEL BARS IN CONTACT WITH STEAM FOR 500 HOURS AT VARIOUS TEMPERATURES

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TABLE II

CORROSION OF STEEL BARS IN CONTACT WITH STEAM AT

Steel	Chemical Analyis (Ladle) Per cent C	Mn	P	S	Si	Cr	Ni	Mo	Cb	Number of Samples	200 Hou A Per
S. A. E. 1010	0.08	0.30	0.017	0.034	—	—	—	—	—	18	0.
Carbon-Moly	0.15	0.44	0.014	0.025	0.17	—	—	0.54	—	—	—
1.25 Cr-Moly	0.11	0.43	0.012	0.012	0.80	1.22	—	0.53	—	6	0.
2 Cr-Moly	0.11	0.47	0.017	0.016	0.40	1.98	—	0.51	—	8	0.
3 Cr-Moly	0.11	0.51	0.014	0.016	0.36	2.95	—	0.98	—	7	0.
4-6 Cr-Moly	0.12	0.41	0.017	0.016	0.28	4.60	—	0.54	—	6	0.
9 Cr-Moly	0.11	0.38	0.010	0.016	0.27	9.00	—	1.22	—	3	0.
12 Cr	0.05	1.19	0.021	0.025	0.38	11.92	—	—	—	8	0.
18-8-Cb stabilized	0.07	0.36	0.015	0.012	0.39	18.62	9.9	—	1.11	8	0.

TABLE III.

STEAM AT 1100 F. FOR 200, 500, 1000, AND 2000 HOURS.

Test							
200 Hour		500 Hour		1000 Hour		2000 Hour	
Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches
18	0.000826	8	0.001238	13	0.001348	16	0.001700
—	———	—	———	—	———	—	———
6	0.000788	4	0.000796	6	———	8	0.001169
8	0.000738	8	0.000811	5	0.001349	8	0.001163
7	0.000748	5	0.000812	8	0.001348	8	0.001131
6	0.000604	—	———	5	0.000943	8	0.000956
3	0.000324	4	0.000459	7	0.000485	4	0.000694
8	0.000095	3	0.000122	8	0.000091	4	0.000045
8	0.000022	4	0.000019	8	0.000079	8	0.000012

TABLE IV.

THE CORROSION OF PLAIN CARBON AND ALLOY STEELS IN CONTACT WITH STEAM BETWEEN 1343 AND 1405 F. FOR 300 HOURS.

Steel	Chemical Analysis (Ladle) Per cent									Temp., F.	Number of Samples	Average Penetration, Inches
	C	Mn	P	S	Si	Cr	Ni	Mo	Cb			
S. A. E. 1010	0.08	0.30	0.017	0.034	.....	.....	.....	.....	.....	1343	4	0.02240
4-6 Cr-Moly	0.12	0.41	0.017	0.016	0.28	4.60	.....	0.54	.....	1357	4	0.01810
7 Cr-Moly	0.11	0.43	0.012	0.011	0.92	7.33	.....	0.59	.....	1371	4	0.00368
9 Cr-Moly	0.11	0.38	0.010	0.016	0.27	9.00	.....	1.22	.....	1382	4	0.00088
12 Cr	0.10	0.51	0.022	0.029	0.40	12.70	.....	.....	.....	1382	4	0.00020
18-8-Cb stabilized	0.08	0.36	0.015	0.012	0.39	18.62	9.90	.....	1.11	1386	4	0.00002
18-8	0.08	1.13	0.014	0.078	0.39	18.79	9.08	0.27	.....	1405	2	0.00002

Seven steels, as listed in Table IV, were subjected to high temperature steam for a period of 300 hours. Heater failures prevented continuation of the test beyond this time. The results are shown in Figure 4 and Table IV. The temperatures at which each steel was tested are shown. Steels having 7 per cent chromium or more showed very little corrosion. The 18-8 stainless steels both stabilized and unstabilized were practically unattacked at these temperatures. The results of this 300-hour test were converted to a 500-hour basis by the use of Figure 2. This extrapolated prediction has been plotted on Figure 3 by dotted lines.

Corrosion of Some Alloy Steels.

A test on the alloy steels shown in Table V was conducted for 500 hours at 1100 F. The results are shown in Figure 5 and Table V. As has been shown in the other tests, the chromium

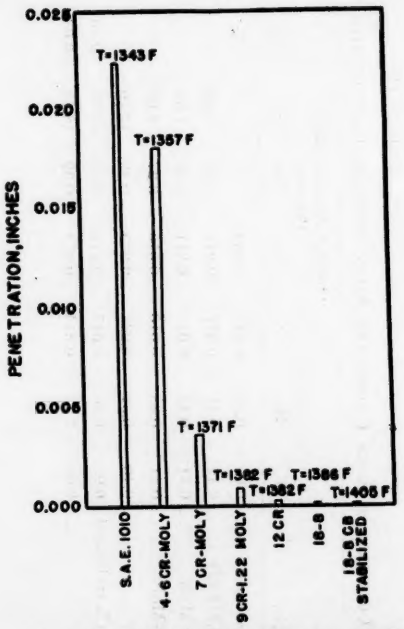

FIGURE 4. CORROSION OF STEELS IN CONTACT WITH STEAM FROM 1343 F TO 1405 F FOR 300 HOURS

TABLE V.

CORROSION OF SOME ALLOY STEELS IN CONTACT WITH STEAM AT 1100 F. FOR 500 HOURS.

Steel	Chemical Analysis (Ladle) Per cent C	Mn	P	S	Si	Cr	Ni	Mo	Al	Cu	Va	Number of Samples	Average Penetration, Inches
S. A. E. 1010	0.08	0.30	0.017	0.034	.....	.....	.....	.....	.....	.....	.....	16	0.001317
Carbon-Moly	0.15	0.44	0.014	0.025	0.17	.....	.....	0.54	.....	.....	.....	16	0.001220
Alloy A*	3.00	1.43	.....	.....	1.72	2.44	14.56	.....	.....	6.39	.....	6	0.001650
Alloy B	0.18	0.52	.....	.....	0.24	1.23	.....	0.23	1.18	.....	.....	6	0.001508
Alloy C	0.07	0.24	.....	.....	1.10	11.57	0.17	1.02	.....	.....	.....	9	0.000076
Alloy D	0.06	0.28	.....	.....	0.34	12.02	0.12	1.16	.....	.....	.....	8	0.000129
Alloy E	0.17	0.50	.....	.....	0.67	1.79	.....	0.69	.....	.....	.....	12	0.001141
Alloy F	0.20	0.46	.....	.....	0.16	0.96	.....	.....	.....	.....	0.15	12	0.001080
Alloy G	0.09	0.32	.....	.....	0.62	0.74	0.31	.....	.....	0.35	.....	11	0.000832
Alloy H	0.11	0.38	.....	.....	0.27	9.00	.....	1.22	.....	.....	.....	11	0.000324

* Iron.

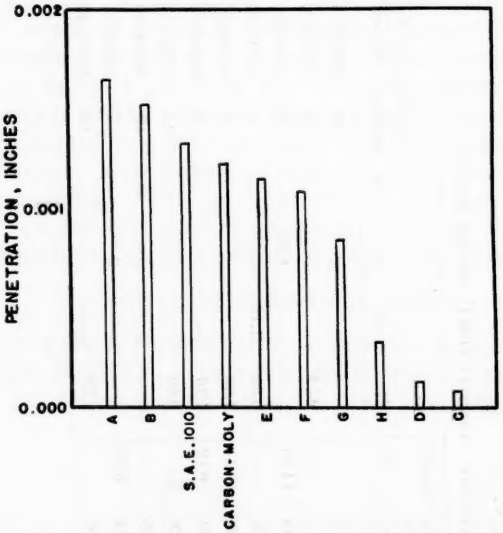

FIGURE 5. CORROSION OF SOME ALLOY STEELS IN CONTACT WITH STEAM AT 1100 F FOR 500 HOURS

content has a marked influence on the corrosion rate. The high copper alloy (A) and the aluminum and low silicon alloy (B) show a higher corrosion rate than either low carbon steel or carbon-moly steel.

Corrosion Tests on Stellite, Hastelloy, Colmonoy and Lamite.

One bar of grade 3 and one of grade 6 Haynes Stellite each 5/8-inch in diameter and 14 inches long were tested in contact with 1100 F. steam for a total of 1300 hours. The cumulative test consisted of two 500-hour, one 200-hour, and one 100-hour tests. Examination of the surfaces after 1300 hours of testing indicates that the amount of oxide formed was too small to be determined by the stripping method. Three specimens each of Hastelloy A, B, and C were placed in contact with 1100 F. steam for a period of 500 hours. A very thin layer of corrosion products

was observed which was too thin to be determined by the stripping operation. Three specimens each of Hastelloy A, B, and C together with pieces of grades 3 and 6 of Haynes Stellite were placed in contact with steam at 1200 F. for 1300 hours. The scale formed was very thin and the amount could not be determined by the stripping process.

Rod samples of Colmonoy and Lamite No. 4, No. 5, and No. 6 were placed in contact with steam at 1200 F. for 570 hours. A very thin layer of scale formed but was too thin to be measured by the electrolytic stripping method. All of these special alloys are extremely resistant to steam corrosion.

Corrosion of Cast Steels.

This test was performed to determine the oxidation of several types of cast steels, the analyses of which are given in Table VI. The test specimens were machined from cast test coupons. The final specimens were 6 inches long and ½-inch in diameter. The specimens were exposed to steam at 1200 F. for 570 hours. Along with the cast steel specimens, samples of 7 Cr rolled steel were

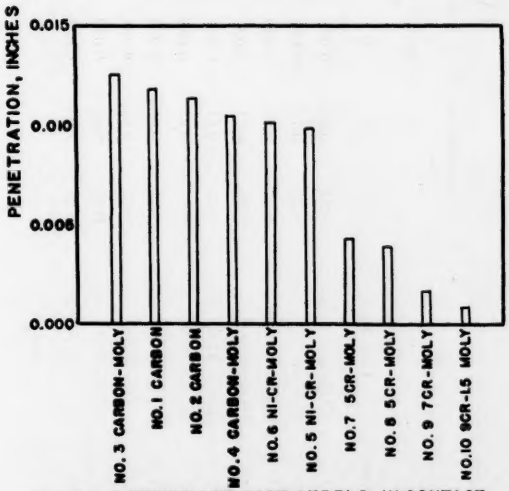

FIGURE 6. CORROSION OF CAST STEELS IN CONTACT WITH STEAM AT 1200 F FOR 570 HOURS

TABLE VI.

CORROSION OF CAST STEELS AT 1200 F. FOR 570 HOURS.

Specimen Number	Type of Cast Steel	Chemical Analysis (Ladle) Per cent C	Mn	P	S	Si	Cr	Ni	Mo	Number of Samples	Average Penetration, Inches
1	Carbon	0.24	0.95	0.020	0.010	0.44	.....	.....	.....	2	0.01180
2	Carbon	0.25	0.75	0.022	0.018	0.44	.....	.....	.....	2	0.01139
3	Carbon-Moly	0.21	0.64	0.020	0.012	0.43	.....	.....	0.49	2	0.01252
4	Carbon-Moly	0.20	0.71	0.020	0.015	0.37	.....	.....	0.49	2	0.01049
5	Ni-Cr-Moly	0.35	0.61	0.021	0.010	0.45	0.64	2.13	0.26	2	0.00992
6	Ni-Cr-Moly	0.28	0.62	0.018	0.010	0.44	0.73	2.25	0.26	2	0.01012
7	5 Cr-Moly	0.22	0.65	0.032	0.010	0.77	5.07	.....	0.47	1	0.00429
8	5 Cr-Moly	0.27	0.63	0.026	0.010	0.87	5.49	.....	0.43	2	0.00395
9	7 Cr-Moly*	0.11	0.43	0.012	0.011	0.92	7.33	.....	0.59	2	0.00173
10	9 Cr-1.5 Moly	0.23	1.05	0.032	0.015	0.84	9.09	.....	1.56	5	0.00088

* Not a cast steel.

TABLE VI

Corrosion of Five Steels by Steam at 1200 F. When Subje

Steel	Chemical Analysis (Ladle) Per cent C	Mn	P	S	Si	Cr	Mo	500 Hour Constant Temperature Number of Samples	Average Penetration, Inches
S. A. E. 1010	0.08	0.30	0.017	0.034	——	——	——	3	0.0105
2 Cr-Moly	0.11	0.47	0.017	0.016	0.40	1.98	0.51	4	0.0094
3 Cr-Moly	0.11	0.51	0.014	0.016	0.36	2.95	0.98	2	0.0092
4-6 Cr-Moly	0.12	0.41	0.017	0.016	0.28	4.60	0.54	3	0.0075
9 Cr-1.22 Moly	0.11	0.38	0.010	0.016	0.27	9.00	1.22	3	0.0015

NOTE: Temperature was constant at 1200 F. for 500 hours, followed by intermittent cooling to room temperatur

TABLE VI

Corrosion of Steels in Contact with Steam During Inte

Steel	C	Mn	P	S	Si	Chemical Analysis Per cent Cr	Ni
S. A. E. 1010	0.15	0.33	0.008	0.028	0.09	0.05	0.08
Carbon-Moly	0.12	0.52	0.010	0.020	0.21	——	——
S. A. E. 6120	0.21	0.43	——	——	——	1.00	——
2 Cr-Moly-Al-Si	0.09	0.28	0.011	0.014	1.27	2.02	——
3 Cr-Moly	0.15	0.48	0.011	0.016	0.45	3.12	——
4-6 Cr-Moly	0.09	0.46	0.030	0.030	0.30	4.75	——
4-6 Cr-Moly-Si	0.13	0.25	0.014	0.012	1.50	4.98	——
4-6 Cr-Moly-Al-Si	0.13	0.34	0.012	0.012	0.82	5.20	——
4-6 Cr-Moly-Cb stabilized	0.10	0.48	0.015	0.014	0.25	6.00	——
4-6 Cr-Moly-Ti stabilized	0.08	0.31	0.015	0.013	0.47	5.18	0.25
9 Cr-1½ Moly	0.13	0.42	0.025	0.016	0.23	8.46	——
12 Cr	0.04	0.75	0.026	0.025	0.31	11.69	0.66
18-8	0.05	0.41	0.020	0.011	0.48	18.23	9.96
18-8-Cb stabilized	0.06	0.58	0.016	0.014	0.38	17.29	12.20

TABLE VII.

When Subjected to Extreme Temperature Fluctuations.

Hour stant erature	Test							
	700 hour Intermittent				1200 Hour Intermittent			
	50 Hour Cycle		100 Hour Cycle		50 Hour Cycle		100 Hour Cycle	
Average Penetration, Inches	Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches	Number of Samples	Average Penetration, Inches
0.0105	3	0.0123	3	0.0113	3	0.0156	4	0.0155
0.0094	2	0.0111	3	0.0105	3	0.0123	3	0.0112
0.0092	4	0.0096	3	0.0089	3	0.0108	3	0.0102
0.0075	3	0.0080	3	0.0084	4	0.0092	3	0.0100
0.0015	3	0.0011	3	0.0011	3	0.0013	4	0.0012

om temperature from 1200 F. after 50 or 100 hour intervals at 1200 F.

TABLE VIII

During Intermittent Operation at 1200 F. for 1300 Hours.

cal Analysis (Ladle) Per cent								
Ni	Mo	Cb	Al	Cu	Ti	Va	Number of Samples	Average Penetration, Inches
0.08	—	—	—	0.014	—	—	4	0.01560
—	0.53	—	—	—	—	—	4	0.01520
—	—	—	—	—	—	0.18	4	0.00809
—	0.55	—	0.70	—	—	—	3	0.00501
—	0.80	—	—	—	—	—	3	0.00755
—	0.55	—	—	—	—	—	3	0.00597
—	0.50	—	—	—	—	—	3	0.00602
—	0.54	—	0.55	—	—	—	3	0.00487
—	0.50	0.77	—	—	—	—	2	0.00487
0.25	0.54	—	—	—	0.63	—	4	0.00433
—	1.52	—	—	—	—	—	3	0.00266
0.66	0.31	—	—	—	—	—	2	0.00128
9.96	—	—	—	—	—	—	4	0.00004
12.20	—	0.68	—	—	—	—	4	0.00003

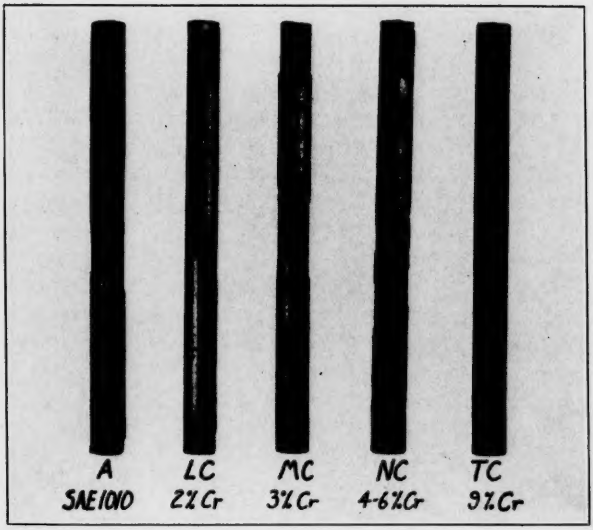

FIGURE 7.—SPECIMENS EXPOSED TO STEAM AT 1200 F. FOR 1200 HOURS. Specimens cooled to room temperature every 50 hours after continuous exposure to steam at 1200 F. for 500 hours.

placed in the reaction chamber for purposes of comparison with earlier results. The results of the test are shown in Table VI and Figure 6. The results from the 7 Cr steel check earlier tests within the accuracy of the experimental measurements. The corrosion decreases as the chromium content increases as was the case with other steels having similar compositions. In general, the corrosion of cast steel specimens is not materially different in amount from the corrosion of rolled steel specimens.

Effect of Temperature Changes on the Corrosion Rate.

This phase of the general test program was undertaken to determine the effect of extreme temperature fluctuations on the corrosion of various steels. Specimens of five different unstressed steels as listed in Table VII were placed in contact with superheated steam at a steady temperature of 1200 F. for 500 hours. At the end of this period three specimens of each steel were removed and stripped electrolytically in order to measure the loss in weight due to corrosion. One-half of the remaining specimens were held at 1200 F. with sudden cooling to room temperature in nitrogen and heating again to 1200 F. every 100 hours until the total time in contact with 1200 F. steam was 1200 hours. The rest of the specimens were treated similarly except for a 50-hour cooling cycle. Specimens were removed after 700 hours and at the end of the test in order to measure the extent of corrosion.

Figure 7 is a photograph of typical specimens at the end of the test. The first visible evidence of extensive cracking of the scale occurred after a total exposure to steam of 700 hours. The result of visual observation during the test indicated that the layer of corrosion products is thinner and more brittle as the chromium content increases up to 5 per cent. An examination of the specimens photographed in Figure 7 shows that the scale on the low carbon steel specimens was very thick, porous, and, in spite of the severe temperature shocks, spalled only slightly as compared to the 4-6 Cr steel. The 2 per cent Cr steel spalled less than the 3 per cent Cr steel while the third layer of scale was cracking in the 4-6 per cent Cr steel at the end of 1200 hours. Little scale formed on the 9 per cent Cr steel and a microscopic examination of the surfaces at the end of the test failed to show evidence that the scale had cracked or checked.

Figure 8 is a plot of the results. It is apparent that the difference in results from the 50 and 100-hour cycle tests is not significant. The thickness of the scale, or in other words, the time of exposure, is the factor which controls cracking and spalling under temperature fluctuations for any given steel. The corrosion decreases as the amount of chromium in the steel increases.

Another test was conducted in order to continue the study on the effect of intermittent heating and cooling of the test samples, and to observe the effect of the specimen surface shape.

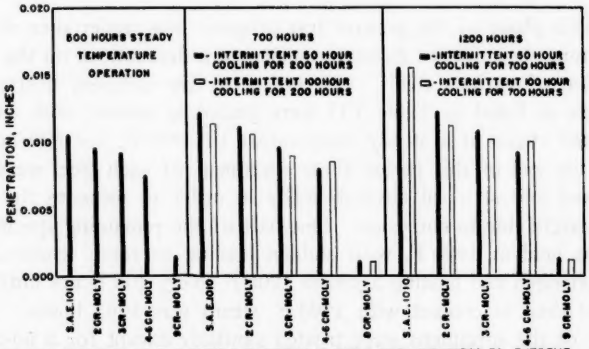

FIGURE 8. CORROSION OF FIVE STEELS BY STEAM AT 1200 F WHEN SUBJECTED TO EXTREME TEMPERATURE FLUCTUATIONS

The ladle analyses of the steels selected for the test are shown in Table VIII. Four inch pieces of 1.5-inch O. D. by 1-inch I. D. tubing were machined externally and internally to remove mill scale, after which hexagonal flats were machined on about 40 per cent of the external surface. The specimens were then split longitudinally. The three types of surface on each piece afforded a means for observing the effect of the shape of surface on the corrosion of the steel.

The specimens were placed in the steam reaction chamber at a temperature of 1200 F. for 500 hours, after which they were removed and rapidly cooled in an atmosphere of nitrogen to room temperature. Two pieces of each steel were removed for use in establishing the oxidation rate for 500 hours. The remaining four pieces of each steel were placed in the reaction chamber at 1200 F. All specimens were removed, cooled to room temperature, examined,

and replaced in the reaction chamber at 600, 700, 800, 900, 1000, and 1150 hours. At the end of 1300 hours of actual test time, the steel samples were removed and the penetration determined. The time intervals previously stated indicate the time during which the specimens were in contact with the steam and do not include the cooling or heating periods.

Before stripping the scale from the steel samples, a 1.25 pound sphere, dropped from a height of 3 feet, struck the end of each piece four times in order to measure the adherent property of the scale.

The scale on all samples was removed at the conclusion of the impact test. After the scale was completely removed, the pieces of steel were weighed in order to determine the loss in weight from which the penetration was computed.

At the end of 500 hours all the steel samples showed a smooth oxide coating without visible cracks in the scale. Several of the samples of 4-6 Cr-Moly-Ti stabilized showed signs of cracking of the scale at the end of 600 hours. This was the first indication of the excessive scaling which was later observed. Samples of 18-8-Cb stabilized and 18-8 unstabilized showed very little scale formation. At the end of 700 hours flakes of scale had chipped off the samples of 4-6 Cr-Moly and 4-6 Cr-Moly-Si. The parent metal which had been exposed by the scale cracks on specimens of 4-6 Cr-Moly-Ti stabilized was again covered with a new layer of oxide. No new changes were observed when the specimens were examined at the end of 800 hours. While the steel samples were being cooled at the 900-hour period, the noise of scale cracking from some of the specimens was audible. The scale on the specimen of S. A. E. 1010 was very thick but adhered rigidly to the metal. The scale on the pieces of 2 Cr-Moly-Al-Si and 4-6 Cr-Moly-Si flaked off in elliptical sections. This type of flaking occurred on the outer curved and flat surface, but did not exist on the inner curved surface. Scale on steels of S. A. E. 6120, 3 Cr-Moly, and Carbon-Moly, although thick, showed no signs of chipping. Complete layers of scale flaked off specimens of S. A. E. 1010, but in no case was the parent metal visible. In all the other steels the parent metal was visible after the scale flaked off. A very thin scale layer formed on steels of 18-8-Cb stabilized, 18-8, 12 Cr, and 9 Cr-1½ Moly. The scale on 4-6 Cr-Moly-Cb stab-

ilized was thicker than on 9 Cr-1½ Moly. Steels of 4-6 Cr-Moly-Ti stabilized 4-6 Cr-Moly, 4-6 Cr-Moly-Si, and 4-6 Cr-Moly-Al-Si all showed sections where large pieces of scale had flaked off. No appreciable changes were noticed at the end of 1150 hours. At the conclusion of the test the scale formed on S. A. E. 1010, Carbon-Moly, 3 Cr-Moly, and S. A. E. 6120 steels was very thick but adhered tightly to the parent metal. Considerable flaking of the scale had occurred on specimens of 4-6 Cr-Moly-Ti stabilized, 4-6 Cr-Moly-Si, 4-6 Cr-Moly, 2 Cr-Moly-Al-Si, and 4-6 Cr-Moly-Al-Si. Very little scale had formed on 18-8-Cb stabilized, 18-8, 12 Cr, and 9 Cr-1½ Moly.

The results of this test have been plotted in Figures 9 and 10. The importance of chromium content on the resistance to corrosion is apparent from these figures. In Figure 10, point (1) is for S. A. E. 6120 which is high in vanadium and point (2) is for 2 Cr-Moly steel high in aluminum and silicon. With the exception of these two points which have been neglected in constructing the curve in Figure 10, the test points for the other steels show a good correlation between chromium content and corrosion penetration.

The steels tested for 1300 hours may be grouped into three general classes according to the scale formed. The first group consists of specimens of Carbon-Moly, 3 Cr-Moly, S. A. E. 6120, and S. A. E. 1010 which were all covered with a thick, porous, tightly adhering scale.

The scale formed on the steels of the second group, 4-6 Cr-Moly-Cb stabilized, 4-6 Cr-Moly-Al-Si, 4-6 Cr-Moly-Si, 2 Cr-Moly-Al-Si, 4-6 Cr-Moly, and 4-6 Cr-Moly-Ti stabilized was very brittle and flaked off the specimens during the test. The scale formed on specimens of 4-6 Cr-Moly-Si and 2 Cr-Moly-Al-Si which were exposed to the steam at 1200 F. for 1300 hours flaked off in elliptical sections. This type of flaking may be due to the silicon in the steel since the silicon contents of 4-6 Cr-Moly-Si and 2 Cr-Moly-Al-Si were 1.50 and 1.27 per cent respectively. Specimens of 4-6 Cr-Moly-Al-Si, which has a silicon content of 0.82 per cent, showed a slight tendency toward this type of flaking. From the visual and microscopic examination of the specimens the scale found on the inside curved surface did not flake off as much as the scale formed on either the outer curved or flat surfaces. No difference in the amount of scale flaking off the outer curved and flat surfaces could be detected.

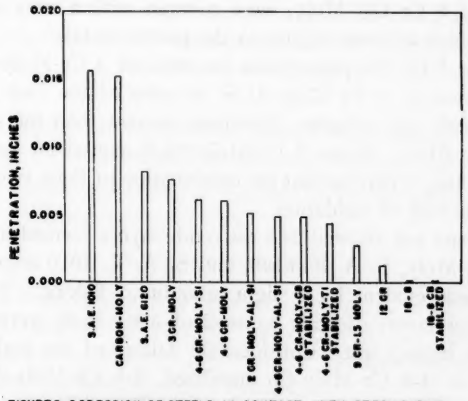

FIGURE 9. CORROSION OF STEELS IN CONTACT WITH STEAM DURING INTERMITTENT OPERATION FOR 1300 HOURS AT 1200 F.

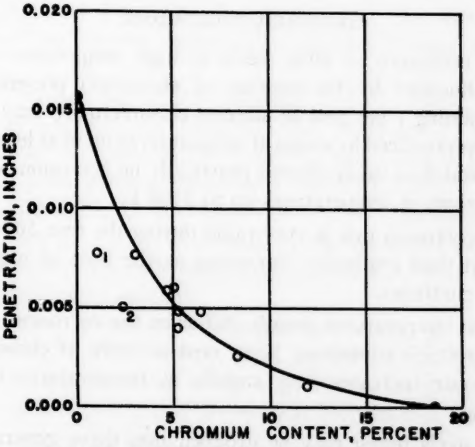

FIGURE 10. THE INFLUENCE OF CHROMIUM CONTENT ON THE CORROSION OF STEELS DURING INTERMITTENT OPERATION AT 1200 F FOR 1300 HOURS.

Steels of the third group consisting of 18-8-Cb stabilized, 18-8, 12 Cr, and 9 Cr-1½ Moly, were covered with a very thin layer of scale which adhered tightly to the parent metal.

In Table VIII the penetration for steel of 3 Cr-Moly is higher than for steel of 2 Cr-Moly-Al-Si in spite of the fact that steel of 3 Cr-Moly has a higher chromium content than the sample of 2 Cr-Moly-Al-Si. Since 2 Cr-Moly-Al-Si contained both silicon and aluminum, it may be that the combination of these two elements reduces the rate of oxidation.

The impact test showed that the oxide layers formed on samples of Carbon-Moly, S. A. E. 6120, and S. A. E. 1010 adhere to the metal as was evident by a slight amount of flaking. Practically all the outer layers of scale on steel of 3 Cr-Moly were removed during the impact test. Considerable flaking of the scale formed on steels of 4-6 Cr-Moly-Cb stabilized, 4-6 Cr-Moly-Al-Si, 4-6 Cr-Moly-Si, 4-6 Cr-Moly, 4-6 Cr-Moly-Ti stabilized, and 2 Cr-Moly-Al-Si occurred during the impact test. No scale was removed from samples of 18-8-Cb stabilized, 18-8, 12 Cr, and 9 Cr-1½-Moly during the impact test.

General Conclusions.

1. The resistance of alloy steels to high temperature steam is greatly influenced by the amount of chromium present. Alloy steels containing 7 per cent or more of chromium are very resistant to corrosion produced by steam at temperatures up to at least 1400 F. The 18-8 stainless steels showed practically no corrosion when subjected to steam at temperatures up to 1400 F.

2. The corrosion rate is very rapid during the first 500 hours of testing and then gradually diminishes as the time of exposure to the steam continues.

3. Steam temperatures greatly influence the corrosion of steels. Except for steels containing 7 per cent or more of chromium the corrosion rate increases very rapidly at temperatures in excess of 1100 F.

4. The steels tested may be grouped into three general classes according to the type of scale formed. The first group consists of low carbon steel, carbon-moly, and the low chromium steels which are covered with a thick, porous, tightly adhering scale. The scale which forms on the steels of the second group, that is the 4-6 Cr

steels and the 2 Cr-Moly-Al-Si steel, is very brittle and easily flakes off under fluctuating temperatures. The third group consists of steels having a chromium content of 7 per cent or more upon which a very thin, non-porous, tightly adhering scale is formed.

5. Scale formed on the inner surface of a tube does not flake off as readily as the scale formed on the outer surface of a tube.

6. Steam pressures between 100 and 1200 psi gauge have no influence on the corrosion of steels.

Acknowledgements.

This investigation has been sponsored by the A. S. M.E. Special Research Committee of Critical Pressure Boilers. The authors wish to express their appreciation of the valuable assistance rendered by the members of this Committee.

In this undertaking Purdue University had effective co-operation from the Engineering Foundation, the American Society of Mechanical Engineers, the Babcock and Wilcox Co., The Detroit Edison Co., The Timken Roller Bearing Co., The Superheater Co., The National Tube Co., The Crane Co., The International Nickel Co., The Haynes Stellite Co., and The Continental Steel Corp.

The authors are particularly indebted to C. L. Clark, research metallurgical engineer, the Timken Roller Bearing Co., Steel and Tube Division, for constructive criticisms and valuable assistance rendered. Special acknowledgement is due H. J. Kerr of the Babcock and Wilcox Co., C. H. Fellows and R. M. Van Duzer of the Detroit Edison Co., E. L. Robinson of the General Electric Co., W. H. Armacost of Combustion Eng. Co., E. C. Petrie and J. J. Kanter of the Crane Company.

To the recent graduates of Purdue University, J. G. Miller, H. Templeton, W. H. Winslow, J. T. Agnew, W. R. Shade, and M. H. Davis the authors owe a debt of gratitude, for it was through their untiring efforts that this project was brought to a satisfactory conclusion.

Bibliography.

1. A. A. Potter, H. L. Solberg, and G. A. Hawkins: Investigation of the Oxidation of Metals by High-Temperature Steam, Trans. A. S. M. E., Vol. 59, No. 8, November, 1937.

THE N. B. T. L. BOILER TEST CODE.

By Fred J. Wiegand.*

One of the primary functions of the Naval Boiler and Turbine Laboratory is to conduct complete operating tests of full size Naval boilers. There are three principal reasons for these tests: (1) To determine whether or not the boiler meets contractual requirements; (2) To obtain data so that instructions can be prescribed for the ship operating personnel; and (3) To obtain data for use in future boiler designs.

None of these objects can be properly evaluated without a complete tabulation of results obtained from calculating the test data. Such a tabulation is commonly known as a long work form. The various items of the combustion calculations and of the heat balance, as well as resultant efficiencies of the boiler, are shown in the long form, or test code. Pertinent information about the boiler and fuel are also given, in addition to the actual data obtained during test.

During the last decade the art of steam generation in Naval vessels has advanced greatly. This progress has largely been due to the proper use of information obtained from results of tests at this Laboratory. As the design of boilers changed, so the number of test code items became greater. A revision and simplification of the code was made in 1936. Again, in 1939, in order to keep abreast of the advance in boiler design, the test code was revised and some of the formula constants were modified slightly to agree with the latest accepted values. Also, further simplification and rearrangement of items were accomplished. It is considered that this code now represents as complete a tabulation of calculated boiler items as is necessary for practical purposes. Although designed primarily for use in conjunction with tests of Naval boilers, it is considered that this form can be used, with only slight modifications, for any type of steam generator installation.

Each modern Naval vessel is provided with a copy of the test report issued by this Laboratory concerning the type boiler installed

* Assistant Mechanical Engineer, Naval Boiler and Turbine Laboratory, Navy Yard, Philadelphia, Pa.

on the ship. The report contains a long work form or test code, with complete calculations and results of the boiler performance. The code, with sample calculations, is shown below.

In view of the fact that all modern Naval vessels are designed for the burning of fuel oil, the Long Work Form and the items contained therein apply to this type fuel.

It is hoped that this presentation will extend the usefulness of these forms which are printed on the following pages.

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA.
BOILER TEST — LONG WORK FORM — SHEET No. 1

Time 12:54-5:00
Date 10-15-35
Test No. 100 *Calculated by* ..... *Run No.* 78

Run No.	ITEM		FORMULA		
	GENERAL DATA				
1	Generating Heating Surface	Sq.Ft.			4,900
2	Superheater Heating Surface	Sq.Ft.			873
3	Economizer Heating Surface	Sq.Ft.			3,900
4	Total Boiler Heating Surface	Sq.Ft.	Item 1 + Item 2 + Item 3		9,273
5	Air Heater Heating Surface	Sq.Ft.			———
6	Total Furnace Volume	Cu.Ft.		Sat. S.H.	580
7	Type of Registers				Standard
8	Opening of Register Doors			Sat. S.H.	Wide Open
9	Position of Burner Barrels	In.		Sat. S.H.	4
10	Type of Sprayers				Mechanical
11	Number and Size of Sprayers			Sat. S.H.	3-4.208
12	Burners in Use			Sat. S.H.	1,2,3
12A	Full Power Equiv. Evap.	Lbs./Hr.			125,040
	FUEL PARTICULARS				
13	N.B.T.L. Number of Oil				1,510
14	Carbon	Percent			87.83
15	Hydrogen	Percent			10.51
16	Sulphur	Percent			0.76
17	Nitrogen	Percent			0.17
18	Oxygen and Undetermined	Percent			0.73
19	Free Moisture	Weight Percent			Trace
20	Heat Value by Calorimeter	BTU/Lb.			18,515
20A	Increase in Heat Value at Fuel Temp. over that at Temp. of Air at Boiler Casing Inlet	BTU/Lb.	0.65 (Item 32 — Item 50)		19
20B	Increase in Heat Value at Constant Press. over that at Constant Volume	BTU/Lb.	$\Delta H_p - \Delta H_v = 25$		25.0
20C	Total Sensible Heat	BTU/Lb.	Item 20 + Item 20A + Item 20B		18,559
	SUMMARIZED TEST DATA				
21	Avg. Carbon Dioxide	Percent CO_2		Sat. S.H.	13.1
22	Avg. Oxygen	Percent O_2		Sat. S.H.	3.9
23	Avg. Carbon Monoxide	Percent CO		Sat. S.H.	0
24	Avg. Nitrogen	Percent N_2		Sat. S.H.	83.0

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA.
BOILER TEST — LONG WORK FORM — SHEET No. 2

Time 12½ X0
Date 10-15-35
Test No. 100 Calculated by ..... Run No. 78

Item No.	ITEM	FORMULA	
28	Avg. Steam Drum Press. Lbs./Sq. In. Abs.		409.8
26	Avg. Steam Drum Outlet Press. Lbs./Sq. In. Abs.		——
27	Avg. S. H. Inlet Pressure Lbs./Sq. In. Abs.		409.2
28	Avg. S. H. Outlet Pressure Lbs./Sq. In. Abs.		408.8
29	Avg. Press. Before Main Stop Lbs./Sq. In. Abs.		408.8
30	Avg. Press. After Main Stop Lbs./Sq. In. Abs.		407.7
30A	Avg. Water Press. Before Econ. Lbs./Sq. In. Abs.	Sat. S.H.	411.8
30B	Avg. Water Press. After Econ. Lbs./Sq. In. Abs.	Sat. S.H.	414.9
31	Avg. Atmospheric Press. Ins. Hg.		30.22
32	Avg. Air Pressure at Boiler Casing Inlet Ins. Water	Sat. S.H.	0.40
33	Avg. Air Pressure at Burners Ins. Water	Sat. S.H.	0.38
34	Avg. Pressure in Furnace Ins. Water	Sat. S.H.	-0.08
35	Avg. Flue Gas Pressure Before Superheater Ins. Water		——
36	Avg. Flue Gas Pressure After Superheater Ins. Water		——
37	Avg. Flue Gas Pressure Before Econ. or Air Heater Ins. Water	Sat. S.H.	-0.10
38	Avg. Flue Gas Pressure After Econ. or Air Heater Ins. Water	Sat. S.H.	-0.09
39	Avg. Pressure in Breeching Ins. Water	Sat. S.H.	——
40	Avg. Pressure at Base of Stack Ins. Water	Sat. S.H.	-0.07
41	Avg. Calorimeter Temperature °F.		317
42	Avg. Steam Temp. at S.H. Outlet °F.		418
43	Avg. Steam Temp. After Main Stop °F.		——
44	Avg. Temp. Feed Water at Feed Stop °F.	Sat. S.H.	190
45	Avg. Temp. Feed Water Leaving Econ. °F.	Sat. S.H.	270
46	Avg. Temp. Gases Leaving G.H.S. °F.	Sat. S.H.	515
47	Avg. Temp. Gases Leaving Boiler °F.	Sat. S.H.	382
48	Avg. Dry Bulb Temp. °F.		89
49	Avg. Wet Bulb Temp. °F.		63
50	Avg. Temp. of Air Entering Boiler Casing Inlet °F.	Sat. S.H.	90
50A	Avg. Air Temp. After Air Heater °F.	Sat. S.H.	——

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA.
BOILER TEST — LONG WORK FORM — SHEET No. 3

Time 12:54-5:00
Date 10-15-35
Test No. 100 Calculated by ..... Run No. 78

Item No.	ITEM	FORMULA		
51	Avg. Air Temp. at Burners °F.		Sat. S.H.	118
52	Avg. Oil Temp. at Burners °F.		Sat. S.H.	132
53	Avg. Oil Pressure at Burners Lbs./Sq. In. Gauge		Sat. S.H.	266
54	Total Oil Burned Lbs.			5,741
55	Total Water Evaporated Lbs.			81,823
56	Duration of Run Hours			4.1

PRIMARY CALCULATIONS

57	Oil Fired Per Hour Lbs./Hr.	Item 54 / Item 56	Total / Sat. S.H.	1,400.2
58	Oil Fired Per Hour Per Sq. Ft. of G.H.S. Lb./Hr./Sq. Ft.	Item 57 / Item 1		0.286
59	Oil Fired Per Hour Per Sq. Ft. of Total Boiler H.S. Lbs./Hr./Sq. Ft.	Item 57 / Item 4		0.151
60	Oil Fired Per Hour Per Sq. Ft. G.H.S. on 18,500 B.T.U. Oil Lbs./Hr./ Sq. Ft.	Item 58 $\left(\frac{\text{Item 20C}}{18,525 + \text{Item 20 A}}\right)$		0.286
61	Oil Fired Per Hour Per Cu. Ft. of Furnace Volume Lbs./Hr./Cu. Ft.	Item 57 / Item 8	Sat. S.H.	2.41
62	Oil Fired Per Burner Per Hour Lb./Hr.	Item 57 / Item 11	Sat. S.H.	466.7
63	Heat Release From Fuel B.T.U. Per Hour Per Cu. Ft. Furnace Volume B.T.U./Hr./Cu. Ft.	Item 61 x Item 20C	Sat. S.H.	44,727
64	Heat in Preheated Air; B.T.U. Per Hour Per Cu. Ft. Furnace Volume B.T.U./Hr./Cu. Ft.	Item 61 (Item 51—100) (0.24 Item 21 + 0.46 Item 22)	Sat. S.H.	273
65	Total Heat Release Per Hour Per Cu. Ft. Furnace Volume B.T.U./Hr./Cu. Ft.	Item 63 + Item 64	Sat. S.H.	45,000
66	Avg. Quality of Steam at S.H. Inlet Percent	$100\left(1 - \frac{0.475\ (T - \text{Item 411})}{L}\right)$		99.8
67	Total Steam Generated Per Hour Lbs./Hr.	Item 55 / Item 56		19,956.8
67A	Sat. Steam Generated Per Hour Lbs./Hr.	From Boiler Data Sheets		4,891.2
67B	S.H. Steam Generated Per Hour Lbs./Hr.	Item 67 — Item 67A		15,065.6
68	Total Heat Absorbed Per Lb. of Sat. Steam B.T.U./Lb.	$\left(\frac{H-h}{h}\right)$ H = Total Heat at Item 26; h = Total Heat at Average Item 44		1,046.5
68A	Total Heat Absorbed Per Lb. of S.H. Steam B.T.U./Lb.	$\left(\frac{H-h}{h}\right)$ H = Total Heat at Item 43, Item 23; h = Total Heat at Average Item 44		1,158.6
69	Heat Absorbed by G.H.S. Per Lb. of Steam B.T.U./Lb.	$\left(\frac{H-h}{h}\right)$ H = Total Heat at Item 26; h = Total Heat at Item 45		965.7
70	Total Equivalent Evaporation Per Hr. Lbs./Hr.	Item 70A + Item 70B		23,264.6
70A	Equivalent Evaporation of Sat. Steam Per Hr. Lbs./Hr.	Item 67A x Item 68 / 970.3		5,275.3
70B	Equivalent Evaporation of S.H. Steam Per Hr. Lbs./Hr.	Item 67B x Item 68 A / 970.3		17,989.3
71	Equivalent Evaporation of G.H.S. Per Hr. Lbs./Hr.	Item 67 x Item 69 / 970.3		19,862.2
72	Total Equivalent Evaporation Per Lb. of Oil Lbs./Lb.	Item 70 / Item 57	Sat. S.H.	16.615
73	Total Equivalent Evaporation Per Lb. of Oil on 18,500 B.T.U. Oil Lbs./Lb.	Item 72 $\left(\frac{18,525 + \text{Item 20A}}{\text{Item 20C}}\right)$	Sat. S.H.	16.602
74	Actual Evaporation Per Lb. of Oil Lbs./Lb.	Item 67 / Item 57		14.253

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA.
BOILER TEST — LONG WORK FORM — SHEET No. 4

Time 12:54-5:00
Test No. 100 Calculated by ..... Date 10-15-35
Run No. 78

Item No.	ITEM		FORMULA		
75	Boiler Horse Power	H.P.	Item 70 / 34.5		674
76	Percent of Full Power	Percent	100 (Item 70 / Item 12A)		18.61
77	B.T.U. Fired Per Hour Per Sq. Ft. G.H.S.	B.T.U./Hr./Sq.Ft.	Item 58 x Item 20C		5,308
77A	Item 77 Based on 18,500 B.T.U. Oil		Item 58 (18,535 + Item 20A)		5,304
78	Equivalent Evaporation Per Hour Per Sq. Ft. of Total Boiler H.S.	Lbs./Hr./Sq.Ft.	Item 79 / Item 4		2.509
79	B.T.U. Absorbed by Total Boiler H.S. Per Hr. Per Sq. Ft.	B.T.U./Hr./Sq.Ft.	Item 78 x 970.3		2,434
80	Actual Evaporation Per Hour Per Sq. Ft. G.H.S.	Lbs./Hr./Sq.Ft.	Item 67 / Item 1		4.07
81	Saturated Steam Temp. at S.H. Outlet Press.	°F.	From Item 28 and Steam Tables		447
82	Superheat at S.H. Outlet	°F.	Item 42 — Item 81		171

COMBUSTION CALCULATIONS

83	Relative Humidity	Percent	From Psychrometric Tables Using Items 43 & 49		24.0
84	Moisture in Air	Lbs./Cu.Ft.	From Psychrometric Tables Using Items 43 & 49		0.0004.82
85	Dew Point	°F.	From Psychrometric Tables Using Items 43 & 49		46.0
86	Vapor Pressure	Ins. Hg.	From Psychrometric Tables Using Item 85		0.3118
87	Weight of Dry Air at Boiler Casing Inlet	Lbs./Cu.Ft.	1.2325 (Item 81 — Item 86 + (Item 82 + 12.5) / 459.6 + Item 50)		0.07227
87A	Item 87 Based on 100° F. Air Temp., Relative Humidity 40%	Lbs./Cu.Ft.	1.2325 (Item 81 — 0.764 + (Item 82 + 12.5) / 559.6)		0.06992
88	Dry Air Theoretically Required for Combustion of 1 Lb. of Fuel	Lbs./Lb.	0.1433 (C/3 + H₂ + S/8 − O₂/8)		13.616
89	Weight of Dry Gases Per Lb. of Fuel	Lbs./Lb.	((4 CO₂ + O₂ + 7N)/3 (CO₂ + CO)) ((C + (8 + 1.286) H₂)/100)	Sat. S.H.	16.982
90	Weight of Moisture from Burning Hydrogen	Lbs./Lb.	Item 15 x 0.0894		0.940
91	Weight of Dry Air Used Per Lb. of Fuel	Lbs./Lb.	Item 89 + Item 90 — ((Item 14 + Item 15 + Item 16)/100)	Sat. S.H.	16.931
92	Weight of Moisture in Air Per Lb. of Fuel	Lbs./Lb.	Item 91 x Item 84 / Item 87	Sat. S.H.	0.113
92A	Item 92 Based on 100° F. Air Temperature, Relative Humidity 40%	Lbs./Lb.	0.00114 Item 91 / Item 87A	Sat. S.H.	0.276
93	Weight of Total Products of Combustion Per Lb. of Fuel	Lbs./Lb.	Item 89 + Item 90 + Item 92	Sat. S.H.	18.035
94	Percentage Excess Air	Percent	100 (Item 91 / Item 88 − 1)	Sat. S.H.	24.35
95	C.F.M. Air Through Blowers	Cu.Ft./Min.	Item 67 / 60 x Item 91 / Item 87	Sat. S.H.	5,447
95A	Air Horsepower	A.H.P.	Item 92 x Item 95 x 0.000156366		0.34
96	Steam Used by Blowers	Lbs./Hr.	From Blower Curves Using Item 95 and Item 82, or Using Item 95A.		190
97	Percent Total Steam Used to Operate Blowers	Percent	100 (Item 96 / Item 70)		0.82
98	Pounds of Steam Used by Blowers per 1000 Cu.Ft. of Air Supplied	Lbs.	16.667 (Item 96 / Item 95)		0.579
99	Percent Total Steam Remaining After Deducting Blower Steam	Percent	100 — Item 97		99.18
100	Lowest Theoretical Stack Temperature	°F.	a. Boiler with Air Heater — Item 50 — Sat. S.H. b. Boiler with Economizer — Item 44 — Sat. Temp. at Item 95 c. Boiler without Air Heater or Econ.		190

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA.
BOILER TEST — LONG WORK FORM — SHEET No. 5

Time 12:5h-5:00
Date 10-15-35
Test No. 100 Calculated by ..... Run No. 78

Item No.	ITEM	FORMULA		
	HEAT BALANCE			
*101	Heat Loss: Moisture from H_2 in Fuel from Temp. Air Entering Boiler Casing to Lowest Theoretical Stack Temp. B.T.U. and Percent	(1089.0 + 0.46 Item 100 — Item 50) Item 90 (See Note)	% B.T.U.	5.50 1,021
*102	Heat Loss: Free Moisture in Fuel from Temp. Oil at Burners to Lowest Theoretical Stack Temp. B.T.U. and Percent	(1089.0 + 0.46 Item 100 — Item 52) $\frac{\text{Item 19}}{100}$ (See Note)	% B.T.U.	0 0
103	Heat Loss: Moisture in Theoretical Air from Temp. Air Entering Boiler Casing to Lowest Theoretical Stack Temp. B.T.U. and Percent	Item 92 x 0.46 (Item 100 — Item 50) $\frac{\text{Item 88}}{\text{Item 91}}$	% B.T.U.	0.02 4
104	Heat Loss: Theoretical Dry Flue Gases from Temp. Air Entering Boiler Casing to Lowest Theoretical Stack Temp. B.T.U. and Percent	0.25 (Item 100 — Item 50) (Item 88 + 1 — Item 90)	% B.T.U.	1.84 342
*105	Total Unavoidable Losses B.T.U. and Percent	Summation Items 101 to 104, inclusive (See Note)	% B.T.U.	7.37 1,367
106	Heat Loss: Incomplete Combustion of Carbon B.T.U. and Percent	$10,160 \left(\frac{CO}{CO_2 + CO}\right)\left(\frac{C}{100}\right)$	% B.T.U.	0 0
107	Heat Loss: Excess Air from Temp. Air Entering Boiler Casing to Lowest Theoretical Stack Temperature B.T.U. and Percent	0.24 (Item 100 — Item 50) (Item 91 — Item 88)	% B.T.U.	0.43 80
108	Heat Loss: Moisture Accompanying Excess Air from Temp. Air Entering Boiler Casing to Lowest Theor. Stack Temp. B.T.U. and Percent	$\frac{\text{Item 103 x Item 94}}{100}$	% B.T.U.	0.01 1
109	Heat Loss: Dry Flue Gases from Lowest Theoretical Stack Temp. to Temp. Gases Leaving Boiler B.T.U. and Percent	0.35 (Item 47 — Item 100) Item 89	% B.T.U.	2.11 391
110	Heat Loss: Moisture in Dry Flue Gases from Lowest Theoretical Stack Temp. to Temp. Gases Leaving Boiler B.T.U. and Percent	0.46 (Item 47 — Item 100) $\left(\text{Item 90} + \text{Item 92} + \frac{\text{Item 19}}{100}\right)$	% B.T.U.	0.24 45
111	Heat Loss: Chargeable to all Absorbing Surfaces B.T.U. and Percent	Item 109 + Item 110 For Unit with A.H.S. Item 105 + Item 109 + Item 110	% B.T.U.	2.35 436
112	Heat Absorbed by Total Boiler Heating Surface B.T.U. and Percent	Item 74 $\left(\frac{\text{Item 68 x Item 67A + Item 68A x Item 67B}}{\text{Item 67}}\right)$	% B.T.U.	86.67 16,122
113	Heat Loss: Unconsumed H_2 and Hydrocarbons, Radiation, and unaccounted for B.T.U. and Percent	Item 20C — (Items 105 + 106 + 107 + 108 + 111 + 112) *Item 105 not to be included for Unit with A.H.S.	% B.T.U.	2.98 553
114	Heat Loss: Chargeable to Furnace and Burners B.T.U. and Percent	Items 106 + 107 + 108 + 113	% B.T.U.	3.42 634
115	Heat Loss: Chargeable to All Absorbing Surfaces and Furnace and Burners B.T.U. and Percent	Item 111 + Item 114	% B.T.U.	5.77 1,070
116	Heat Absorbed by G.H.S. Per Lb. of Fuel B.T.U. and Percent	Item 69 x Item 74	% B.T.U.	7.42 13,764
117	Heat Absorbed by E.H.S. Per Lb. of Fuel B.T.U. and Percent	Item 74 (h₁ — h₂) h_1 = Heat of Liquid at Item 45 h_2 = Heat of Liquid at Item 44	% B.T.U.	6.21 1,152
118	Heat Absorbed by S.H.S. Per Lb. of Fuel B.T.U. and Percent	Item 112 — (Item 116 + Item 117)	% B.T.U.	6.50 1,206
119	Heat Absorbed by Air from Entrance to Fireroom to Boiler Casing Inlet Per Lb. of Fuel B.T.U. and Percent	(0.24 Item 91 + 0.46 Item 92) (Item 50 — Item 48)	% B.T.U.	0.04 8
119A	Heat absorbed by Air from Boiler Casing Inlet to Burners Per Lb. of Fuel B.T.U. and Percent	(0.24 Item 91 + 0.46 Item 92) (Item 51 — Item 50)	% B.T.U.	0.62 115
120	Heat Available for Absorption by Total Boiler Heating Surface B.T.U. and Percent	Item 111 + Item 112	% B.T.U.	89.22 16,558

*NOTE: For Unit with A.H.S., these losses are avoidable, since stack temperature may be same as inlet air temperature. These Items are only to be used for inclusion in Total Heat Losses.

NAVAL BOILER LABORATORY, NAVY YARD, PHILADELPHIA, PA. Time 12:54-5:00
BOILER TEST — LONG WORK FORM — SHEET No. 6 Date 10-15-35

Test No. 109 *Calculated by* ..... Run No. 78

Item No.	ITEM	FORMULA		
121	Heat Available for Absorption by E.H.S. B.T.U. and Percent	Item 111 + Item 117	% B.T.U.	8.56 1,508
122	Heat Available for Absorption by A.H.S. B.T.U. and Percent	Item 111 + Item 119A	% B.T.U.	——
	RESULTANT EFFICIENCIES			
123	Overall Efficiency of Total Boiler H. S. and Furnace and Burners Percent	Item 113 / Item 20C		86.87
124	Combined Efficiency of Absorbing Surfaces Percent	Item 112 / Item 120		97.37
125	Combined Efficiency of Furnace and Burners Percent	Item 120 / Item 20C — Item 105	For unit with A.H.S. Item 120 / Item 20C	96.31
126	Combined Efficiency of Absorbing Surfaces and Furnace and Burners, eliminating unavoidable losses Percent	Item 112 / Item 20C — Item 105	This Formula does not apply for unit with A.H.S.	93.78
127	Efficiency of G.H.S., S.H.S., and Furnace and Burners, without E.H.S. or A.H.S. Percent	Item 112 — Item 117 / Item 20C	This Formula does not apply for unit with A.H.S.	80.66
128	Efficiency of G.H.S. and S.H.S. Percent	Item 116 + Item 118 / Item 120		90.41
129	Efficiency of E.H.S. Percent	Item 117 / Item 121		72.54
130	Efficiency of A.H.S. Percent	Item 119A / Item 122		——
131	Fireroom Efficiency Percent	$97.030 \left(\text{Item } 72 - \frac{\text{Item } 96}{\text{Item } 97} \right) \div \text{Item } 20C$		86.16
	PERTINENT PRESSURE LOSSES			
132	Pressure Loss: Boiler Casing Inlet to Burner Front Ins. Water	Item 32 — Item 33	Sat. S.H.	0.02
133	Pressure Loss: Through Registers Ins. Water	Item 33 — Item 34	Sat. S.H.	0.46
134	Pressure Loss: Through S.H.S. Ins. Water	Item 35 — Item 36		——
135	Pressure Loss: Through G.H.S. and S.H.S. Ins. Water	Item 34 — Item 37	Sat. S.H.	0.02
136	Pressure Loss: Through E.H.S. or A.H.S. (Gas Side) Ins. Water	Item 37 — Item 38	Sat. S.H.	-0.01
137	Pressure Loss: Steam Drum to S.H. Inlet Lbs./Sq. In.	Item 25 — Item 27		0.6
138	Pressure Loss: Through Superheater Lbs./Sq. In.	Item 27 — Item 28		0.4
139	Pressure Loss: Through Main Stop Lbs./Sq. In.	Item 29 — Item 30		1.1
140	Pressure Loss: Steam Drum to After Main Stop Lbs./Sq In	Item 25 — Item 30		2.1
140A	Water Pressure Loss: Through Economiser Lbs./Sq. In.	Item 30A — (Item 30B — Item 140C)	Sat. S.H.	0.2
140 B	Vertical Height Between Econ. Inlet and Econ. Outlet Ft.			8.33
140 C	Static Head of Water from Econ. Inlet to Econ. Outlet-Lbs/Sq. In.	Item 140B x 0.3981		3.3

Although it is considered that no difficulty will be experienced by the user of this boiler test code, a short discussion concerning a few of the items is given below for additional clarity.

Item 20-B—This item is of greater magnitude with gaseous fuel than with liquid fuels and it is possible in some instances that this value may be sufficiently small in the case of the fuel oil to be within the error encountered in the determination of the heating value of the oil.

Items 29, 30 and *43* are values which allow the determination of line losses, after the boiler outlet, if so desired, and are not to be used in the determination of the boiler heat balance.

Item 66—In this formula, T is the calorimeter temperature when the steam is known to be of 100 per cent quality and L is the latent heat of vaporization. The values of both T and L, of course, must correspond to the line pressure at the calorimeter. T is usually determined for each boiler over the range of working steam pressures and a calibration curve drawn. For any particular test run, T can be read from the curve and L can be obtained from steam tables.

Item 87—The constant, 1.3268, is obtained as follows:

$PV = WRT \qquad \frac{W}{V} = \frac{P}{RT}$ Since the weight per cubic foot is desired, V becomes unity.

P (lbs. per sq. ft.) $= \frac{144(p)}{2.036} = 70.7269(p)$, where p is in inches of mercury.

$R = \frac{1544}{\text{molecular weight}}$. The average percentages of O_2, N_2, and A in dry air are 20.99, 78.06, and 0.95, respectively. Therefore, the molecular weight equals $0.2099 \times 32 + 0.7806 \times 28.016 + 0.0095 \times 39.944 = 28.9656$. Then $R = \frac{1544}{28.9656} = 53.305$ and

$$W = \frac{P}{RT} = \frac{70.7269(p)}{53.305(T)} = 1.3268\left(\frac{p}{T}\right).$$

Item 88—The constant 0.3422 is in reality 34.22 which has been divided by 100 so that the percentage values of C, H_2, S, and O_2 as

recorded on sheet 1 of the code may be used directly in the formula. The International Critical Tables show the density of oxygen to be 1.429 grains per liter and of air 1.293 grains per liter while the volume percentage of oxygen in air is given as 20.99. The percentage by weight, then, is $\frac{20.99 \times 1.429}{1.293} = 23.20$. Each pound of O_2 thus requires $\frac{100}{23.20} = 4.31$ pounds of air and, since 7.94 pounds of O_2 are required for each pound of H_2, the theoretical air required for each pound of H_2 is therefore $4.31 \times 7.94 = 34.22$. Since an equal weight of carbon combines with only one-third as much air as does hydrogen, the percentage of carbon in the fuel is divided by 3 in this formula. Similarly, the sulphur percentage is divided by 8. The percentage of oxygen can be subtracted directly from that of sulphur because the molecular weight of S is practically the same as that of O_2.

Item 89—The first factor of this formula will be recognized more readily by the reader when written $\frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}$. In order to employ a short-cut method of calculating this item, we have simplified this factor by substituting for N_2 its equivalent in terms of the other variables. Therefore,

$$\frac{11CO_2 + 8O_2 + 7[CO + (100 - CO_2 - O_2 - CO)]}{3(CO_2 + CO)} =$$

$$\frac{11CO_2 + 8O_2 + 7CO + 700 - 7CO_2 - 7O_2 - 7CO}{3(CO_2 + CO)} =$$

$$\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}.$$

It will be noted that the second factor of this formula contains a correction for sulphur. Its value is obtained in the following manner: Using atomic weights of 12 for C, 32 for O_2, and 32.064 for S, one pound of C will burn to 3.667 pounds of CO_2 and one pound of S will produce 1.998 pounds of SO_2. The ratio of these weights $= \frac{3.667}{1.998} = 1.8353$. The percentage of sulphur in the

fuel should be divided by this ratio as shown by Item 89 of the code.

Item 90—The constant 0.0894 is really 8.94 which has been divided by 100 so that the percentage volume of H as recorded on sheet 1 may be used directly. Using the atomic weight of H as 1.0078 and of oxygen as 16.0000, $\frac{16+2(1.0078)}{2(1.0078)}=\frac{18.0156}{2.0156}=8.94$.

Item 95A—This formula is derived as follows:

$$\text{Air Horsepower}=\frac{\text{ft.-lbs. per minute}}{33{,}000}=\frac{\frac{h}{12}(62.0\ \text{CFM})}{33{,}000}$$

$$=0.000156566(h)(\text{CFM}),$$

where h = air pressure at boiler casing inlet = Item 32.

CFM = air supplied by blowers = Item 95.

The weight of water at 100 degrees F. (62.0 pounds per cubic foot) is used because this temperature of air at the boiler casing entrance is the present design value.

Item 98—The quantity of steam used by the blowers to supply one cubic foot of air equals the steam used to supply the total quantity of air multiplied by the density of air divided by the weight of air supplied. The formula to obtain the pounds of steam used to supply 1000 cubic feet of air then becomes $\frac{\text{Item 96}\times\text{Item 87}\times 1000}{\text{Item 91}\times\text{Item 57}}$.

This can be simplified by multiplying both numerator and denominator by 60 and separating the formula into two factors as follows: $\left(\frac{\text{Item 96}\times 1000}{60}\right)\left(\frac{\text{Item 87}\times 60}{\text{Item 91}\times\text{Item 57}}\right)$. The second factor is the reciprocal of Item 95. Therefore, the first factor can be divided by Item 95 instead of being multiplied by the second factor. The formula then becomes $\frac{\text{Item 96}\times 1000}{\text{Item 95}\times 60}$ which equals

$$16.667\left(\frac{\text{Item 96}}{\text{Item 95}}\right).$$

Item 140C—The constant 0.3981, representing the pressure of one foot of water in pounds per square inch, is based on water at 300 degrees F. This is the average temperature of water in the economizers of recent Naval boilers.

The boiler test code discussed above, although generally similar in content to that recommended by the American Society of Mechanical Engineers, is unique in its method of treating the various factors of a boiler test. It will be recognized that the code makes no correction to data as entered, i. e., all primary data from test are to represent true values when entered on the form. The subject of taking boiler test data and making corrections thereto is beyond the scope of the present monograph. It is hoped that this subject can be fully presented by one of the Laboratory's staff in some future issue of the JOURNAL.

THE INFLUENCE OF CORROSION ON PROPELLER SHAFT MAINTENANCE.

By H. L. Setz.*

In light of our present national emergency, we realize the prime necessity of utilizing every available means to coordinate our maintenance program to meet performance standards of our equipment with the least possible additional load on repair and replacement facilities.

With the work loads of propulsion and auxiliary machinery manufacturers increasing daily, the procurement of spare parts presents a problem of growing complexity. This is especially true in respect to parts such as propeller shafts, whose large proportions require relatively long periods of time to complete the manufacturing operations. As facilities for this type of work are limited, serious delays in delivery may be experienced should an emergency arise, requiring extensive and immediate replacements.

With these factors in mind, the following discussion is presented relating corrosion to serviceability of propeller shafting.

It is generally understood that surface conditions have a very definite effect on the fatigue properties of a material. In a cylindrically shaped member, torsionally loaded, such as a propeller shaft where the maximum stress occurs at the outer surface, the surface condition is a factor of major importance. The presence in varying degrees of torsional vibration only serves to make this condition more predominant.

It is well known that any sudden variation in crosssection, surface conditions, or continuity of material, is responsible for high localized stresses. Corrosive action produces transformations in the material surface; for example, pitting, though infinitesimal in nature when in the primary stages, serves as seats from which further corrosive action develops. The resulting irregular skin

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condition is responsible for high localized stresses occurring in areas immediately surrounding individual discontinuities. This condition of localized stress concentration, in conjunction with cyclic stress resulting from torsional vibration, is responsible for the formation of a miniature crack, or cracks, when such stress exceeds the strength of the material. This pitted or cracked condition of the material surface, when a direct result of corrosion, may be considered the first stage of corrosion fatigue.

If at this time all corrosive influences were removed from the vicinity of the surface, the member would ultimately fail due to crack propagation or spreading. This second stage is essentially a fatigue stage, with the failure proceeding according to the general laws of fatigue. The predominant feature of this phenomenon is the propagation, under applied cyclic stresses, of a crack by stress concentrating effects, the magnitude of which is primarily controlled by the shape of the crack and the physical properties of the material.

Several methods of preventing corrosion of propeller shafting are in general use. They consist of coating the shafting with non-corrosive compounds among which are zinc and rubber. Even with these various precautions, corrosion sometimes occurs when sea water gains access to the shaft surface beneath the protective coating.

The procedure suggested requires the removal of the shaft to some convenient location where satisfactory machine shop facilities are available. In order to eliminate all traces of corrosion from the surface of the shaft, it may be necessary to reduce its diameter an amount ranging between a few hundredths to perhaps even ½ an inch, depending on the seriousness of the corrosion and the size of the shaft. The resulting new smooth surface eliminates those discontinuities which are responsible for inducing high localized stresses which may result eventually in ultimate shaft failure.

When reducing the shaft diameter to eliminate corrosion, great care should be exercised to leave no abrupt changes in cross-section, as they are responsible for high stress concentrations. At all times very gradual changes in cross-section and generous fillets should be used.

When corrosion occurs on shaft surfaces, within stuffing boxes and bearings, additional problems arise. When shaft diameters are

reduced in these localities, minor changes in design of stuffing boxes and bearings may be required. When corrosion is discovered within the bearings, a thorough investigation of the bearing lubrication system should be undertaken and, whenever possible, defects eliminated.

After smooth surface conditions have been restored to the shaft, further precautionary measures should be exercised to prevent continued corrosive action on the shaft surface. This may be accomplished by again applying a protective coating of a suitable non-corrosive substance. The manner of application of such a substance should be supervised with due care and diligence, so that the resulting bond will be an effective seal against corrosion producing elements.

If corrosive action is arrested in time, subsequent failures may be avoided and shaft life and serviceability extended considerably. It is also believed that cost and time factors involved in the above mentioned procedure are small in comparison to those accompanied by new shafting replacements.

It is difficult to determine the exact magnitude of the effect of corrosion upon the fatigue properties of the material. It is believed that, in cases of excessive surface discontinuity, the fatigue strength of the shaft may be reduced as much as 40 or 50 percent. In such cases, shaft failures occur as a result of average stress values somewhat smaller than those which are considered safe operating stresses under normal conditions.

The point which may prove the greatest bone of contention is the question of how much the cross-sectional area of the shaft may be reduced, in order to eliminate corroded surfaces without impairing its overall strength. It is a matter of choice between reducing the designed strength of the shafting by reducing the cross-sectional area and improving the fatigue properties by eliminating corroded surfaces. These factors are interdependent to an extent depending on the relative seriousness of the corrosion condition and to what extent the area of cross-section must be reduced to eliminate corroded surfaces.

Due consideration should also be given changes in the vibrational characteristics of the shaft resulting from the reduction of its area of cross-section. Such reductions of area would materially affect the shaft's natural frequency, amplitude of vibration, position of nodes, critical speed and hysteresis damping characteristics.

The magnitude and nature of these changes are dependent on the quantity of material removed from the shaft and the distribution along the length of the shaft of the material removed. In practice, the final decision on the individual case should rest with a competent engineer who is acquainted with the factors involved.

In order to present a clearer and more exacting picture of the problem, computations of average stress values and factors of safety for a series of solid shafts, ranging in diameter from 3 to 12 inches, have been undertaken. The diameters of these same shafts were then reduced ⅛ inch, ¼ inch, and ½ inch, respectively, and new values of average stress and factors of safety were computed, assuming the same loading conditions as in the first case. The results obtained give some indication of the effect of reducing the cross-sectional area of a shaft upon its overall strength when subjected to identical operating conditions.

For illustrative purposes, an example was taken from actual operating conditions, namely, the port inboard propeller shaft of the U. S. S. *Pennsylvania,* which failed in 1932 as a result of simultaneous action of fatigue and corrosion.

The following table presents the results of chemical analysis obtained from the subject shaft. All values are reported in percent and represent the average of at least two determinations. Requirements of Navy Department Specifications 49S2f for Class An nickel steel are included. The values listed below are quoted directly from the report on the *Pennsylvania* shaft.

Material	C	S	P	Mn	Si	Ni	Cr
Subject Shaft	.39	.035	.038	.51	.09	3.10	.13
Navy Dept. Spec. 49S2f	Max.	Max.	Max.			Min.	
Class An nickel steel	.45	.045	.040			3.00	

As observed from these tables, the steel meets the requirements of Specifications 49S2f, with the exception of "yield point" which was slightly low as determined by relatively accurate means. The report of the laboratory which examined the failed shaft also states that the "yield point" value, as determined by the drop of the beam of the testing machine, would probably be within the requirements of the specifications.

TENSION TEST RESULTS.

Material Direction	Tensile Strength	Johnsons Limit	Proof Stress	Elastic Limit	Proportional Limit	Elong. in. 2 in.	Reduction of Area
	Lb.	per Sq.	In.			Percent	
Subject Shaft Longitudinal	87100	48000	47500	47000	46600	28.7	51.1
Shaft Transverse	86000	41200			38000	14.7	19.5
Navy Dept. Spec. 49S2f	80000	50000x				L 25.0 T 21.0	L 45.0 T 30.0

SHEAR, CHARPY IMPACT, AND HARDNESS TEST RESULTS.

Shearing Strength 16 per sq. in.	Charpy Impact Value, ft. lb.	*Brinell	Hardness †Rockwell "B" Scale
L 67900	L 16.1	172	B 86.1
T 68000	T 14.8		

* 3000-Kg. load, 10-MM diameter ball
† 100-kg. load, 1/16 inch diameter ball
L Longitudinal direction
T Transverse direction
X yield point

An examination of the microstructure of the steel indicated that the section had been cooled slowly through the critical temperature range, a normal condition for this type of material.

From these observations, in addition to a visual examination of the nature of the failure, it was concluded that failure of the shaft was caused by corrosion simultaneous with fatigue.

The following computations are presented using as a hypothesis actual operating conditions of the propeller shaft of the U. S. S. *Pennsylvania* referenced above, and certain assumptions stated below.

Propeller shaft dimensions:

Inside diameter = 7.25 inches

Outside diameter = 12.25 inches

Shaft horsepower (1 shaft) = 8000 at 215 RPM. for full power run.

Then from the fundamental horsepower-torque relation,

H.P. $= \frac{2\pi NT}{33000}$ we can solve for the torque developed at full power

$$T = \frac{H.P. \times 33000}{2\pi N}$$

Where T = torque expressed in foot pounds

H. P. = full power — foot pounds per minute

N = RPM.

Substituting the above values in the relation we have

$$T = \frac{8000 \times 33000}{2\pi \times 215} = 196000 \text{ foot pounds}$$

Then utilizing the fundamental shear stress relation, $S_s = \frac{Tr}{J}$ for a torsionally loaded member we may now calculate the working stress for full power conditions.

$$S_s = \frac{Tr}{J} = \frac{196000 \times 12 \times 6.125}{1935} = 7450 \text{ pounds per square inch}$$

where S_s = shear stress induced in the outer surface of the shaft expressed in pounds per square inch.

T = torque expressed in inch pounds.

r = the distance from the center of gravity to the most remote fiber,—inches

J = the polar moment of inertia of the cross-section expressed in $in.^4$

For a hollow shaft $J = \frac{\pi}{32}(D^4 - d^4)$

$$= \frac{\pi}{32}(12.25)^4 - (7.25)^4 = 1935 \text{ } in.^4$$

Where D = outside diameter of the shaft—inches

d = inside diameter of the shaft—inches

The factor of safety may now be computed from the shear strength and the full power working stress S_s

$$\text{F.S.} = \frac{\text{Shear strength — pounds per square inch}}{S_s \text{ — pounds per square inch}}$$

$$= \frac{68000}{7450} = 9.13$$

Assuming that the material of the solid shaft series (3 inches diameter to 12 inches diameter) is identical with that of the preceding example and that the allowable working stress is the same, equivalent torque loading may be computed for each individual shaft.

$$S_s = \frac{Tr}{J} \text{ or } T = \frac{S_s J}{r}$$

$$= \frac{7450\ J}{r}$$

This relation may be solved for equivalent torque loading values by substituting values of J and corresponding to shaft diameters 3, 4, 5, 6, 7, 8, 9, 10, 11, and 12 inches respectively, as indicated in columns A, E, and H of Shaft Computations.

Each of shafts of the subject series is reduced in diameter ⅛ inch, ¼ inch, and ½ inch respectively, as would be the case when eliminating corroded surfaces after assuming a condition of loading identical with that of each shaft size prior to reduction of its diameter. The resulting working stresses and factors of safety are computed as illustrated below.

$$S_s = \frac{Tr}{J} = \frac{39500 \times 1.4325}{6.70} = 8440 \text{ pounds per square inch}$$

$$\text{F.S.} = \frac{68000}{8440} = 8.05$$

These results are obtained with the use of values of J and r corresponding to the reduced diameters as indicated in columns F, H, and I of Shaft Computations.

The percent reduction in factor of safety, resulting from reduction of shaft diameter, may now be computed.

$$\text{Percent reduction F.S.} = \frac{(\text{original F.S.}) - (\text{reduced F.S.}) \times 100}{\text{Original F. S.}}$$

$$= \frac{(9.13 - 8.05)}{9.13} \times 100 = 11.83 \text{ percent}$$

These results are shown in column L of Shaft Computations.

All computations are carried out to normal slide rule accuracy.

In order to more accurately ascertain the quantitative effects of corrosion on the fatigue limit of steels, reference is made to experimental data published in Mechanical Engineering, November 1938,

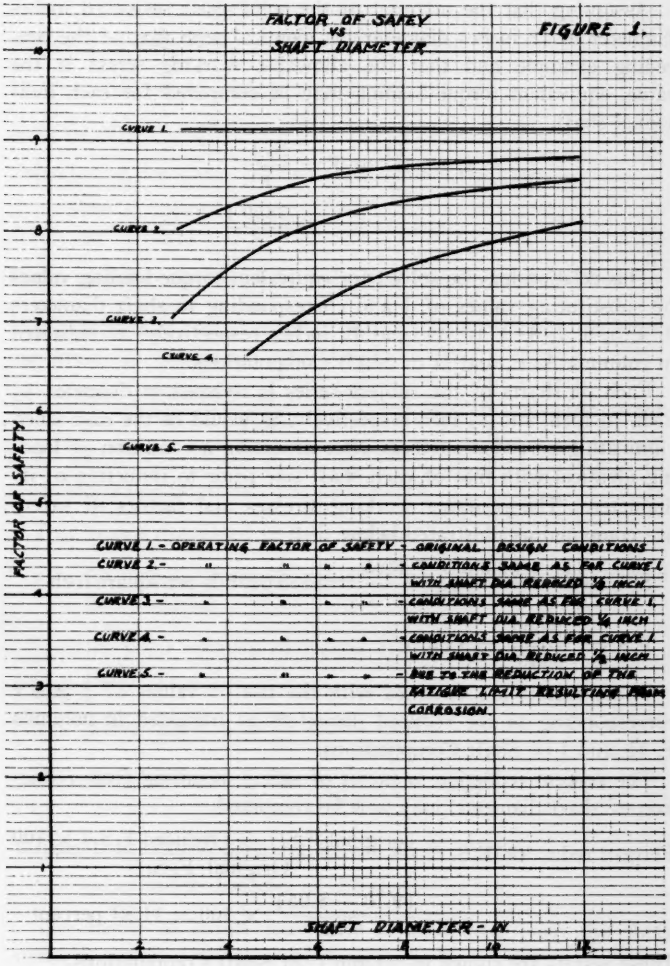

FACTOR OF SAFEY
VS
SHAFT DIAMETER
FIGURE 1.
CURVE 1.
CURVE 2.
CURVE 3.
CURVE 4.
CURVE 5.
FACTOR OF SAFETY
CURVE 1 - OPERATING FACTOR OF SAFETY - ORIGINAL DESIGN CONDITIONS
CURVE 2 - " " " " - CONDITIONS SAME AS FOR CURVE 1 WITH SHAFT DIA. REDUCED 1/8 INCH
CURVE 3 - " " " " - CONDITIONS SAME AS FOR CURVE 1, WITH SHAFT DIA. REDUCED 1/4 INCH
CURVE 4 - " " " " - CONDITIONS SAME AS FOR CURVE 1 WITH SHAFT DIA. REDUCED 1/2 INCH
CURVE 5 - " " " " - DUE TO THE REDUCTION OF THE FATIGUE LIMIT RESULTING FROM CORROSION.
SHAFT DIAMETER - IN
2
4
6
8
10
12

Table 1—

A	B	C	D	E	F
Original Shaft Dia. Inches	Reduced Shaft Dia. Inches	Original Shaft Area In.^{2}	Reduced Shaft Area In.^{2}	Original $J = \frac{\pi R^4}{32}$ In.^{4}	Reduced $J = \frac{\pi R^4}{32}$ In.^{4}
3	2⅞	7.07	6.48	7.95	6.70
3	2¾	7.07	5.95	7.95	5.64
4	3⅞	12.57	11.78	25.15	22.10
4	3¾	12.57	11.04	25.15	19.40
5	4⅞	19.65	18.65	61.40	55.50
5	4¾	19.65	17.75	61.40	50.1
5	4½	19.65	15.90	61.40	40.3
6	5⅞	28.3	27.15	127.20	117.2
6	5¾	28.3	26.00	127.20	107.5
6	5½	28.3	23.80	127.20	90.0
7	6⅞	38.5	37.10	236.0	219.0
7	6¾	38.5	35.80	236.0	204.0
7	6½	38.5	33.20	236.0	175.2
8	7⅞	50.3	48.80	402.0	378
8	7¾	50.3	47.20	402.0	355
8	7½	50.3	44.20	402.0	311
9	8⅞	63.6	61.80	644	608
9	8¾	63.6	60.10	644	575
9	8½	63.6	56.70	644	512
10	9⅞	78.6	76.50	982	932
10	9¾	78.6	74.60	982	886
10	9½	78.6	70.90	982	800
11	10⅞	95.1	92.8	1438	1370
11	10¾	95.1	90.8	1438	1313
11	10½	95.1	86.7	1438	1197
12	11⅞	113.2	110.7	2038	1950
12	11¾	113.2	108.6	2038	1872
12	11½	113.2	103.9	2038	1718

Table 1—SHAFT COMPUTATIONS.

F	G	H	I	J	K	L
Reduced $\frac{J = \pi R^4}{32}$ In.^{4}	Full Load Max. Power Working Stress #1 In.^{2}	Equivalent Torque Acting on Original Diameters In. lb	New Increased Working Stress Due to Decreased Shaft Dia. #1 In.^{2}	Original Factor of Safety	Reduced Factor of Safety	% Reduction in Factor of Safety %
6.70	7450	39,500	8440	9.13	8.05	11.83
5.64	7450	39,500	9630	9.13	7.06	22.65
23.10	7450	93,600	8220	9.13	8.27	9.42
19.40	7450	93,600	9070	9.13	7.50	17.86
55.50	7450	183,000	8050	9.13	8.45	7.45
50.1	7450	183,000	8680	9.13	7.84	14.13
40.3	7450	183,000	10220	9.13	6.66	27.05
117.2	7450	316,000	7930	9.13	8.58	6.03
107.5	7450	316,000	8450	9.13	8.05	11.83
90.0	7450	316,000	9670	9.13	7.03	23.00
219.0	7450	503,000	7890	9.13	8.62	5.48
204.0	7450	503,000	8310	9.13	8.19	10.30
175.2	7450	503,000	9330	9.13	7.30	20.05
378	7450	748,000	7790	9.13	8.73	4.38
355	7450	748,000	8160	9.13	8.33	8.77
311	7450	748,000	9020	9.13	7.54	17.42
608	7450	1,067,000	7780	9.13	8.74	4.28
575	7450	1,067,000	8110	9.13	8.39	8.10
512	7450	1,067,000	8860	9.13	7.68	15.90
932	7450	1,462,000	7750	9.13	8.78	3.84
886	7450	1,462,000	8040	9.13	8.46	7.34
800	7450	1,462,000	8680	9.13	7.83	14.23
1370	7450	1,947,000	7720	9.13	8.81	3.50
1313	7450	1,947,000	7970	9.13	8.53	6.58
1197	7450	1,947,000	8540	9.13	7.96	12.80
1950	7450	2,527,000	7690	9.13	8.84	3.18
1872	7450	2,527,000	7940	9.13	8.57	6.13
1718	7450	2,527,000	8460	9.13	8.04	11.95

Table 2—CORROSION-FATIGUE LIM

MATERIAL	Tensile Strength lb. per sq. in.
SAE 1035	88,500
SAE 1050	93,400
SAE 1050 water-quenched and drawn	130,100
SAE 2315	79,700
SAE 2335 normalized	104,300
SAE 3130 normalized	100,500
SAE 4130 water-quenched and drawn	128,500
SAE 4615 normalized	91,300
SAE 9260 normalized	143,800
Pearlitic manganese normalized	118,200
5 percent chrome, oil-quenched and drawn	130,600
Nickel-Copper, normalized	113,000
Copper-nickel	77,300
Wrought Iron	47,700
3.5 percent nickel wrought iron	57,400
3 percent nickel molybdenum iron	63,600

IGUE LIMITS OF STEELS IN SALT WATER IN ABSENCE OF AIR.

e Strength per sq. in.	Fatigue limit pounds per square inch			Fatigue Ratio	Damage Ratio	
	Air	Corrosion	Sulphide Corrosion		Corrosion	Sulphide Corrosion
88,500	40,600	24,600	10,600	0.46	0.61	0.26
93,400	31,600	19,900	10,900	0.34	0.63	0.34
30,100	60,100	25,400	13,900	0.47	0.42	0.23
79,700	51,900	31,600	23,900	0.65	0.61	0.46
04,300	53,900	39,900	24,900	0.52	0.74	0.46
00,500	55,100	31,600	15,900	0.55	0.57	0.29
28,500	70,100	26,900	14,100	0.55	0.38	0.20
91,300	48,600	33,100	22,400	0.53	0.68	0.46
43,800	72,100	24,900	14,900	0.50	0.35	0.21
18,200	56,400	29,400	12,100	0.48	0.52	0.21
30,600	73,900	52,900	15,500	0.57	0.72	0.21
13,000	58,900	26,100	21,900	0.52	0.44	0.37
77,300	54,100	33,600	19,600	0.70	0.62	0.36
47,700	30,400	19,600	16,400	0.64	0.64	0.54
57,400	39,600	26,900	19,100	0.69	0.68	0.48
63,600	45,100	25,400	21,900	0.71	0.56	0.49

under the title "Fatigue and Corrosion Fatigue." The results of corrosion fatigue tests in absence of air are given in Table 2. The tests were made on rotating-beam at 1750 cycles per minute for 10,000,000 cycles of stress. The water was a brine containing 67,940 ppm of sodium chloride. This brine was saturated with hydrogen sulphide for the sulphide corrosion-fatigue tests. The damage ratios are taken as the ratio of corrosion-fatigue limit to fatigue limit, and sulphide-corrosion-fatigue limit to fatigue limit, respectively.

Of the materials referenced in Table 2, steels SAE 1035, SAE 1050, and SAE 2315 will with least error conform to the Navy specifications for propeller shaft material. Using the values of damage ratio for these materials, it is now possible to make a graphical comparison between reduction in shaft strength or factor of safety resulting from reducing shaft diameters without altering loading conditions and the reduction of shaft strength due to corrosion fatigue. This comparison is illustrated in Figure 1 by plotting the factors of safety resulting from the various conditions, against shaft diameters.

Curve 5, Figure 1, was obtained from the damage ratios listed in Table 2 for the three steels referenced above. It represents the effect of corrosion on the fatigue limit of these steels when subjected to the action of a sodium chloride solution containing 67,940 ppm of sodium chloride in the absence of air. As illustrated in Figure 1, this result is expressed in terms of the factor of safety, because any reduction in the fatigue limit of a material as a result of corrosion or for other reasons will affect a corresponding reduction in the operating factor of safety.

Curve 5, Figure 1, was obtained in the following manner: Averaging damage ratios of the 3 steels referenced above and listed in Table 2, in order to obtain a single result.

We have: $\frac{.61 + .61 + .63}{3} = \frac{1.85}{3} = .6167$ average damage ratio.

The reduced factor of safety as a result of corrosion = (original factor of safety = 9.13) $\times$ .6167 = 5.62

This value is plotted as curve 5.

It is pointed out that the salt solution, used in the tests from which the data in Table 2 was taken, was of higher concentration than sea water, the corrosive agent attacking propeller shafts. The higher salt concentration used for the tests may result in more pronounced and accelerated corrosive action, decreasing to some degree and damage ratios which would have resulted if sea water had been used. The experimental work was conducted in the absence of air, while in actual operating conditions air may be present, thus accelerating corrosion. It is believed that the overall result simulated in curve 5 is within reasonable accuracy.

Curve 1 represents the operating factor of safety as computed under original conditions of design and loading.

Curve 2 illustrates the effect of reducing the shaft diameter ⅛ inch on the operating factor of safety, assuming the same loading conditions as in curve one.

Curves 3 and 4 demonstrate the effect of reducing shaft diameters ¼ inch and ½ inch respectively, other factors remaining the same as in the case of curve 2.

Curves 2, 3, and 4 also conclude that the detrimental effect of reducing all shaft diameters an equal amount decreases as individual shaft diameters increase.

A factor of safety of 7 is generally considered fairly safe for the operation of propeller shafts. From this it may be concluded that any reduction in shaft diameter to eliminate corroded surfaces is to be recommended, provided the resulting factor of safety is in excess of 7. As previously stated, the decision of individual cases should rest in the hands of a competent engineer.

Due to the lack of experimental facilities, it was necessary to take data from recorded operating conditions and other experimental work. With the aid of this data and various assumptions, a theoretical analysis was developed. While the results are not intended to serve as absolute quantitative criteria, it is hoped that the graphical comparison of Figure 1 will give the reader a definite insight as to the relative impetus of the factors involved.

FUEL INJECTION PUMPS FOR DIESEL POWERED NAVAL MOTOR BOATS.

By Mr. Carroll R. Alden.*

It may truly be said that the fuel injection pump is the heart of the Diesel engine. This statement applies particularly to high speed light weight Diesels in which the time available for the injection is inversely proportional to the engine speed.

It has long since been demonstrated that fuel pumps respectively built in accordance with several different schools of thought relative to their underlying principles of operation, will each successfully operate the several types of engines to which they have been correctly applied, often with a close agreement in the matter of specific fuel consumption and specific power ouput. It therefore logically follows that questions of preferment of the various designs rests, not in theoretical considerations but upon the suitability of the respective designs to the particular conditions of use and service. This paper therefore deals with practical rather than theoretical aspects of fuel injection pumps for Naval motor boats.

It is well known that larger Diesels suitable for powering of cargo and submarine craft reached a stage of practicability a considerable number of years in advance of the smaller, light weight, high speed type required for the powering of Naval motor boats.

The Bureau of Engineering of the United States Navy had much to do with the development of these smaller Diesels. This is clearly indicated by the fact that, for many years prior to commercial use of engines of this type for any purpose whatever, the U. S. Naval Experiment Station at Annapolis was performing exhaustive tests on those sample engines which were submitted by engine builders in the hope that their product would meet the exacting requirements of the Navy.

*Research Engineer of Ex-Cell-O Corporation.

The weaknesses discovered in the tests of these early engines had much to do with instituting the improvements which were necessary before such engines could be successfully used for motor truck, industrial or privately owned power boats. American engine builders have the U. S. Naval experiment station to thank for insistance upon a power/weight ratio, specific fuel consumption, durability, reliability, cold starting ability and general utility rating for these small engines, lacking which they would not have been acceptable for the many non-military uses which they are now successfully serving.

The reasons for the interest of the Navy Department in Diesel engines as replacements for the gasoline engines which were then currently used in Naval motor boats are many. Most important of course is the increased safety on ship board which obtains when gasoline can be eliminated as an item of ship's stores.

Prior to 1935 the Buda Company of Harvey, Illinois submitted for test at Annapolis, three sizes of engines which, from a mechanical standpoint, met the requirements of the Bureau of Engineering as of that date. These engines had one important disadvantage for Naval use. The only fuel injection pumps which could then (1935) be purchased for such engines were of German design and were *made in Germany.* (Pumps of the design last referred to are now currently made in the United States.)

Some time before this, development had begun on the first fuel injection pump of American manufacture to be offered to American engine builders. See Figures 1, outboard view and 2, inboard view. Testing of this pump was promptly started at Annapolis and was successfully completed in time that the first fleet installations of Dieselized motor boats in the Navy were equipped with this product of American design and manufacture.

The German pump design herein referred to comprises vertical plungers and barrels, one for each engine cylinder, in an en bloc mounting which is supported on a bracket or sub base which must be provided alongside the engine. A cam shaft and tappets gives motion to the respective pump plungers, said shaft being driven through an adjustable slip coupling if fixed or invariable injection timing is acceptable or must be driven through oppositely and spirally splined shafts adapted for variation of their phase relationship by manual shifting of a sliding coupling if variable injection timing is desired.

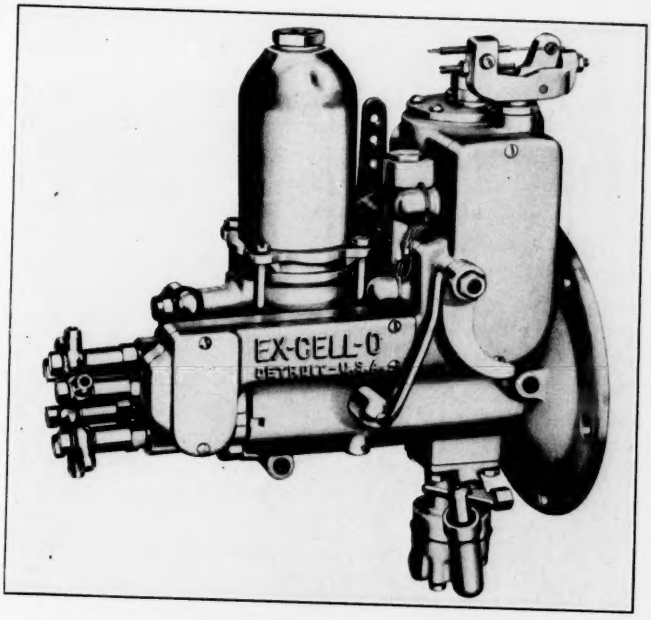


FIGURE 1.

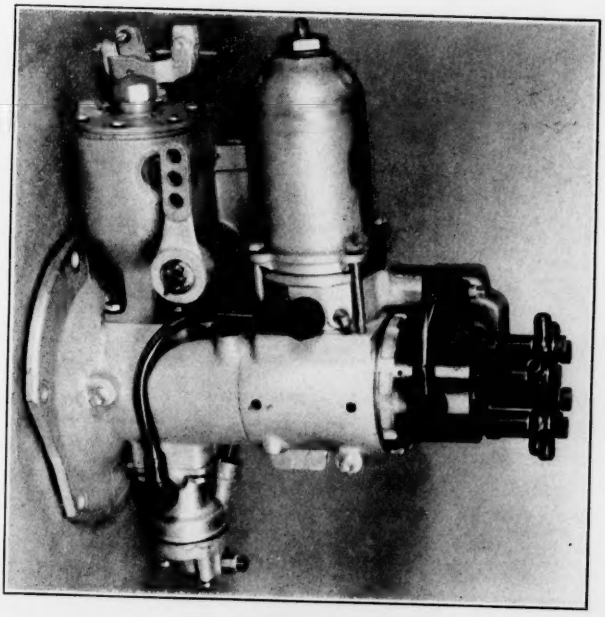


FIGURE 2.

In no case was automatic alteration of injection timing with change in engine speed, made available and in no case was it possible to avoid the transmission of the rather severe shocks and impulses in the driving torque of the fuel pump cam shaft through the timing control device if variable timing were elected.

Replacement of fuel pump barrels and plungers in the motorboat or even on board ship or at repair stations was impracticable to the point of virtual impossibility as ensuing adjustment of the relative injection timing and quantity of fuel deliverey to the respective engine cylinders requires that the pump be removed from the engine and adjustment of these functions be accomplished with the aid of a testing stand on which timing and quantity of fuel delivery can be determined while adjustment is being made.

A number of other important points in design will occur to the reader as the discussion of the American pump then chosen by the Navy Department for this duty, proceeds.

The pump described is called an American design not only because it was designed in America by Americans but because it incorporates American principles of manufacture, use and servicing as illustrated by the following features which first made their appearance anywhere in the world, in this fuel injection pump.

All moving parts are protected from reach and hidden from view—this makes for safety.

Flange mounting, gasket tight, on the engine, makes for cleanliness.

Being gear driven without slip or demountable coupling makes for quietness.

Pressure lubrication from the engine lubricating oil system makes for certainty of lubrication and length of life of plunger actuating parts.

Incorporation of a fuel safety filter as a part of the pump assembly makes for length of life of the parts handling fuel under high pressures.

Designing the pump as an assembly of replaceable sub assembly units permits maximum rapidity of repairs by exchange of units. This is the servicing plan which had brought the automobile to its present state of utility. Replacement of pumping parts can be made in the motor boat if necessary and in any case without dis-

mounting the fuel pump proper and without subsequent timing and fuel quantity adjustments to separate pump cylinders.

Designing the pump with a choice of available strokes through choice of swash plate angles as will later appear, permits the precision hydraulic pumping parts to be made with one plunger diameter which is capable of serving a wide range of engine sizes. This reduces the number of plunger sizes to *one* rather than a different plunger size for each engine size.

Designing one replaceable hydraulic unit containing all fuel handling parts *and no adjustments* prevents derangement of the pump by alteration of the arrangement of its parts either through curiosity, ignorance or malice.

Incorporating in the pump assembly an automatic float controlled drip catcher which returns leakage fuel from pump, nozzles and air vent to the fuel supply source saves fuel, prevents smell in the boats and saves painted surfaces.

Designing the pump for automatic advance of timing with increase in engine speed results in quieter engine idling, better fuel consumption and less exhaust smoke at top speed full load.

Designing the pump for mass production by American manufacturing methods means that manufacturing capacity can be rapidly expanded to meet war time volume without excessive dependence upon highly skilled manual labor.

Navy motor boat engines, models DA, DB and DD, following closely the original Buda design but much improved in power output, largely as a result of the cooperation of Naval engineers at the Norfolk Navy Yard, have been manufactured at that station since shortly after the purchase of the first Diesels from the Buda Company.

Figure 3 shows a partial horizontal cross section through the drive unit and both horizontal and vertical transverse cross sections through the hydraulic unit.

On the horizontal cross section is seen the primary fuel connection carrying filtered fuel from the discharge side of the fuel safety filter to the inlet of the hydraulic unit, the central rotary valve with its driving connection and the fuel quantity control cross shaft by which endwise motion is communicated to the central rotary valve for the purpose of controlling the fuel quantity delivered by the pump.

In the vertical cross section the six high pressure pump plungers are seen symmetrically and hexagonally arranged around and parallel to the central rotary valve.

The nut by which the hydraulic unit is mounted is also visible in the horizontal section. This was also shown in Figure. 3.

Figure 3 also shows the fuel quantity control cross shaft and its connecting clevis and the spring case by which the plungers and

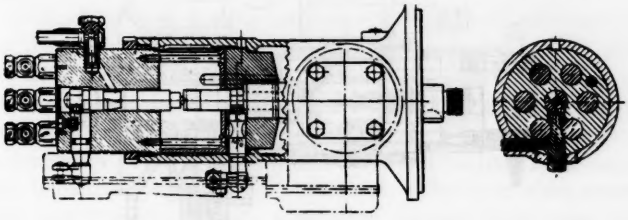


FIGURE 3.

their return springs and spring hangers are prevented from separating from the hydraulic unit stator or body part when the hydraulic unit is dismounted from the drive unit.

Figure 4 shows the mounting and driving of the transfer pump and the drive to the rotary valve of the hydraulic unit.

A main fuel pump drive shaft extends axially through the center of the drive unit. The pump driving gear is mounted by means of a keyed taper and a lock nut on the forward end of the drive shaft. The back end of this shaft has an axial bore traversed by a transverse cross pin which is slightly off center. A spirally slotted timing member having an outside diameter sufficiently small to allow it to slide axially is contained in this bore. The spiral slots allow the timing member to slide over the transverse cross pin when presented in the proper phase relation but prevent engagement when incorrectly presented.

The back end of the timing member is transversely slotted slightly off center to engage a driving tang by which the central valve of the hydraulic unit may be turned at pump shaft speed which is equal to the cam shaft speed of the engine.

As before the off center location of the slot in the timing member and the tang in the rotor driver prevent their being engaged in any but the correct driving relationship.

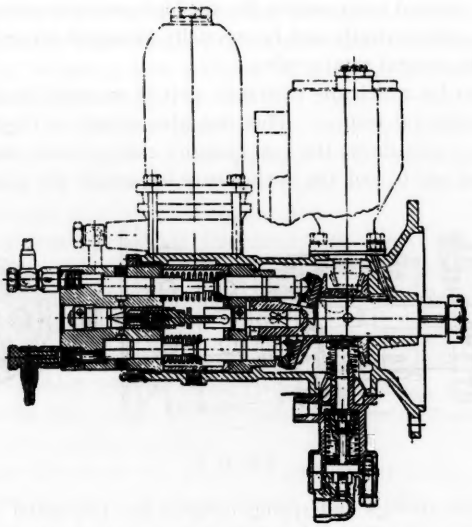


FIGURE 4.

A timing control cross shaft about midway of the length of the drive unit, in a horizontal axial plane and running to the outboard side of the drive unit, has on its inboard end an eccentric ball ended pin which engages a horse shoe washer surrounding a shouldered neck portion of the timing member about midway of its length.

Rocking of the timing control cross shaft causes the timing control member to be displaced axially which, because of its spiral slots, causes the phase of the central rotary valve to be shifted relative to the main pump drive shaft.

The timing control cross shaft has at its outboard end a follower in a plane at right angles to the cross shaft. The follower is made of about 5/16 rod stock and is urged by torsion spring into engagement with a sickle cam which is a part of the engine speed control throttle system later to be described. Thus with each movement of the speed throttle into a higher speed position the timing control cross shaft is caused to move the timing control member into a position which advances the phase of the central rotary valve

of the hydraulic unit relative to the drive shaft of the pump. This advances the timing of the injection relative to the engine piston in a manner which will be described later.

Surrounding the pump drive shaft and firmly fixed thereto by longitudinal key and transverse pin will be found a member which has on one end a centrally facing bevel gear for driving a vertical governor shaft. The back of this gear is formed with a thrust face.

Centrally of this member and immediately below the driver gear on the governor shaft is found an eccentric portion which gives stroke to the tappet of the fuel transfer pump.

Oppositely disposed with respect to its mid length, this member has a planar swash plate surface which forms an acute angle with a transverse plane, both planes however intersecting along one of the diameters lying in a transverse plane. Therefore when rotated, this swash plate, through a properly disposed shoe plate with loosely held shoes, is capable of giving longitudinal reciprocatory motion to six tappets which are respectively aligned with the pump plungers of the hydraulic unit. The stroke of these tappets and plungers is dependent upon the angle of the swash plate and the radius of the tappet circle. The swash plate angle is chosen to give to the plungers, which are of one given diameter, regardless of engine size, a choice of length of strokes which determines the possible pump plunger displacement in accordance with the engine cylinder displacement.

As the angle of the swash plate in any given pump or for any given engine is a fixed quantity it is necessary to change the quantity of pump delivery in accordance with engine speed and load conditions by an entirely different method which will now be described.

Surrounding the innermost ends of the plungers of the hydraulic unit and extending axially of the same for a distance somewhat greater than their maximum possible travel is a counterbore of slightly greater diameter than the plungers.

By-pass cross bores drilled in a plane transverse to the plungers and along lines joining the center lines of opposite plungers, connect these counterbored clearance spaces with the bore in which the central rotary valve is rotatable and axially shiftable.

This central rotary valve is formed somewhat like a spool having cylindrical portions on either end which make a pressure tight journal in the bore which surrounds the valve.

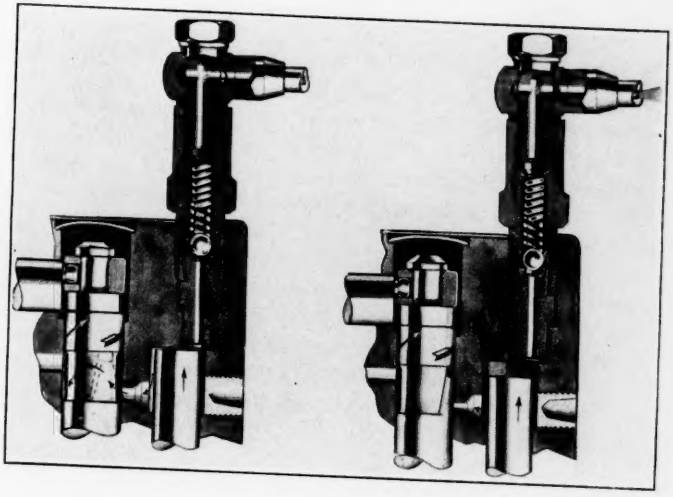
A central portion of the spool between these end cylindrical portions is reduced in diameter save for a triangular land portion having for its altitude an extended element of one of the cylindrical ends.

At its apex this triangular land portion is not sufficiently wide to completely close the by pass cross holes over which it passes in its rotation but when axially shifted so that its base portion passes successively over the cross holes of the separate pump cylinders, each of these holes is successively closed for an angular duration of pump shaft rotation equal to the maximum duration of fuel delivery in degrees of crank shaft travel which is desired of the fuel pump.

When less than the maximum duration (and quantity) of fuel delivery is desired, axial movement of the central rotary valve caused by rocking movement of the cross shaft, eccentric pin and horse shoe collar surrounding a neck portion on the end of the valve (essentially similar to the construction by which the timing member may be axially shifted) makes it possible to control the quantity (and duration) of delivery. This rocking motion is given to the fuel quantity control cross shaft by a governor later to be described.

It will be recalled that the fuel is conducted through a drilled passage in the wall of the stator, or body of the hydraulic unit and, at all times, fills the cavity formed between the valve bore and the reduced diameter of the spool-shaped rotary valve. So long as the by-pass ports leading from this cavity to the counterbore surrounding the plungers, are not covered by the land on the rotary valve, the fuel displaced by movement of each plunger is free to pass into or out of the respective by-pass port.

Figure 5 comprises four views of the relations between a given plunger of the pump and the rotary valve in different phases of the pump cycle. (View 1) The rotation of the valve is so timed in relation to the reciprocation of the plungers that an excess of fuel is drawn into the respective plunger cavities upon the outward or filling stroke of the plungers. (View 2) After the plunger has started its inward or expulsion stroke, fuel is returned to the rotary valve cavity until the plunger has reached the desired velocity in



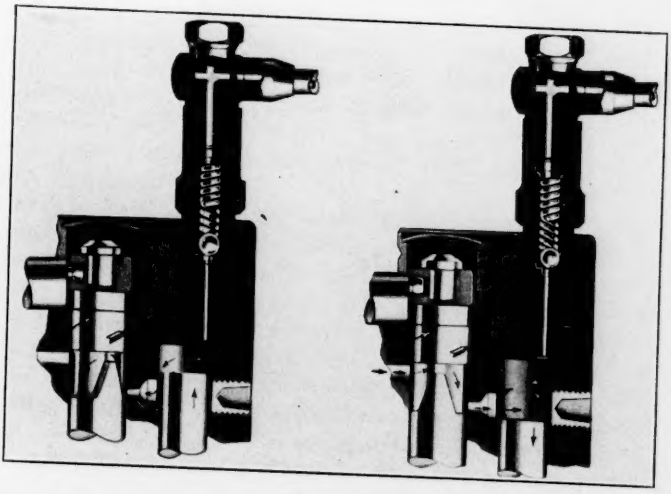


FIGURE 5.

the discharge direction. (View 3) At this time the plunger is about mid-way of its inward stroke and the valve has rotated to a point where the leading edge of the triangular land on the valve just covers the opening into the valve cavity which is made by the by-pass port. Further inward travel of the plunger causes a pressure to be built up in the counterbored plunger cavity. This pressure quickly becomes high enough to lift the associated fuel injection valve against its calibrated opening pressure, after first having opened the discharge check valve of the respective pump cylinder. The closing of the by-pass port in the manner described, is timed not only with the reciprocation of the fuel pump plungers but with the piston and valves of the respective engine cylinders. (View 4) After opening of the by-pass valve the inward stroke of the plunger is completed under conditions substantially as described re View 1.

It is now seen that advance of the angular phase relation of the rotating valve with respect to the reciprocation of the pump plungers (and therefore with respect to the injection top center of the engine pistons) advances the timing of the injection.

It is particularly to be noted that the drive to the pump plungers is not affected by change in the timing adjustment. It is highly advantageous that change of timing be possible without altering the phase relation between the engine and the pump drive shaft. In the latter case the timing control mechanism is forced to withstand the shocks and impulses which must be transmitted therethrough. These are quite severe in the case of cam type pump drives yet there is no alternative but to cause the phase change mechanism to transmit the driving impulses when a variable timing of such pumps is necessary.

Because of the fact that the variable timing device of the pump is not required to transmit the driving torque for the entire pump with its shocks and impulses, the mechanism can be much lighter and less expensive and requires much less controlling force for its actuation. This readily permits Automatic Timing Control.

Figure 6. In this view is seen the vertical governor shaft which is bevel gear driven from the main pump shaft and which rotates a pair of flyball governor weights pivoted to give change of position to a spool member which slides vertically on the vertical shaft and, in moving, gives a change of position to a fuel quantity control

arm which is enclosed with connecting linkage, in a housing on the side of the governor casing and fuel pump drive unit.

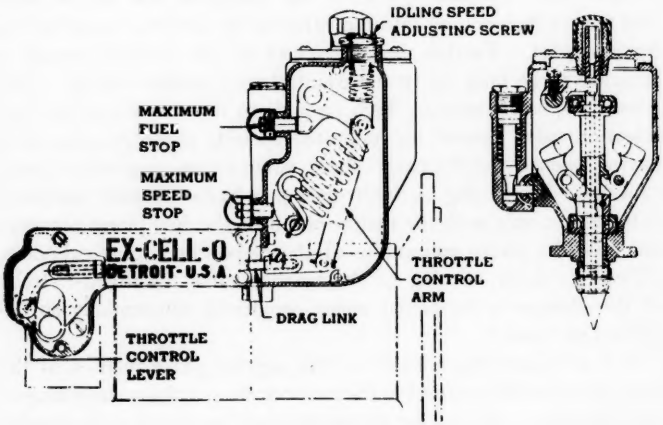


FIGURE 6.

The linkage includes a speed control spring which may be tensioned by a lever externally positioned and joined to the engine speed control throttle. When this throttle is carried through the slow and idling speed positions and forced against the resistance of a spring pressed idling restraining plunger into the stop position, the fuel quantity control arm is moved by compressive forces transmitted through the now collapsed speed control spring, into the zero delivery or "stop" position. In all of its postions the movements of the free end of the quantity control arm are transmitted through a drag link to the offset clevis of the fuel quantity control cross shaft of the hydraulic unit. The action in controlling the quantity of fuel delivered from the hydraulic units has already been described.

When the external speed control throttle is placed in the position which neither tensions nor compresses the speed control spring, a smaller compression spring applies to the quantity control arm of the governor that amount of force which is required to balance the forces set up by the flyballs at idling speed and causes an idling quantity of fuel to be delivered to the engine.

The sickle cam which has previously been described as altering the injection timing is carried on the same shaft, remotely actuated by the operator's speed throttle, which causes increased tension on the governor speed control spring when increased engine speed is desired. Therefore an advance in engine timing occurs with each increase in tension placed by the operator's speed control throttle upon the speed control spring.

The governor has three adjustable limit stops, two of which may be capped and sealed after adjustment by an authorized person. The two stops which may be sealed after adjustment are the maximum fuel quantity stop which limits the travel of the fuel quantity control arm in the direction of increased fuel quantity and the maximum speed stop which limits the movement of the speed control throttle in the direction which increases tension on the governor speed control spring. The first of these determines the maximum fuel quantity which can be delivered, thus limiting the maximum torque which may be delivered by the engine. The second limits the tension which can be placed on the speed control spring and therefore limits the engine speed irrespective of the engine torque output (so long as this speed is not limited by the load applied to the engine).

The stop which requires no seal is the adjustment by which the normal strength of the idling spring may be adjusted. As this adjustment only determines the speed at which the engine will idle, no damage will result to the engine if the adjustment be varied between quite wide limits and no seal on this adjustment is necessary.

Other than those previously mentioned, there are no adjustments on the Navy pumps except those which affect the length of the drag link, these being set and locked at the factory. It is thought to be a matter of note that the hydraulic unit, that small but important package which contains all of the precision parts handling the fuel just prior to injection, can be made without any adjustable parts whatever.

Absence of adjustments is especially important in equipment used by the combatant services as alteration through curiosity, ignorance or malice is rendered impossible.

EFFICIENCY OF DOUBLE REDUCTION GEARS AS INFLUENCED BY LUBRICATING OIL TEMPERATURES.

By R. Michel, Member.

The employment of higher turbine wheel speeds in recent naval construction has resulted in the almost universal use of double reduction gears for geared turbine drives. The wheel speed of a turbine, which is defined as the linear velocity in feet per second of the mean blade diameter, is limited by the allowable centrifugal stress in the roots of the blades of the last stage. Improvements in blading materials and methods of construction have raised wheel speeds of marine turbines to about 800-900 feet per second. In order to realize high efficiency in the turbine over its whole range of operation, the ratio of wheel speed to steam speed must not vary greatly from a constant which depends upon the type of turbine. For a Rateau wheel, this constant, corresponding to maximum engine efficiency, has a theoretical value of 0.47. Its value is double this for reaction blading.

High propeller speeds are not conducive to high propeller efficiencies. The use of a single reduction gear necessitates too large a bull gear diameter if the propeller and turbine are both to operate at high efficiency. Double reduction gears have therefore been used on all recent naval vessels.

The fact is overlooked, however, that the efficiency of the double reduction gear, while only slightly under that of the single reduction gear at full power operation, is considerably less than that of the single reduction gear at cruising speeds. Typical efficiency curves for destroyer single and double reduction gears supplied with Navy Symbol 2190 lubricating oil at 110 degrees F. are shown in Figure 1, the variation of efficiency with power output being shown. It will be noted that the efficiencies of single and double reduction gears at full power are 98.3 per cent and 97 per cent, respectively, whereas at 1/20th of full power these are approximately 97 per cent and 91.6 per cent, respectively. A considerable

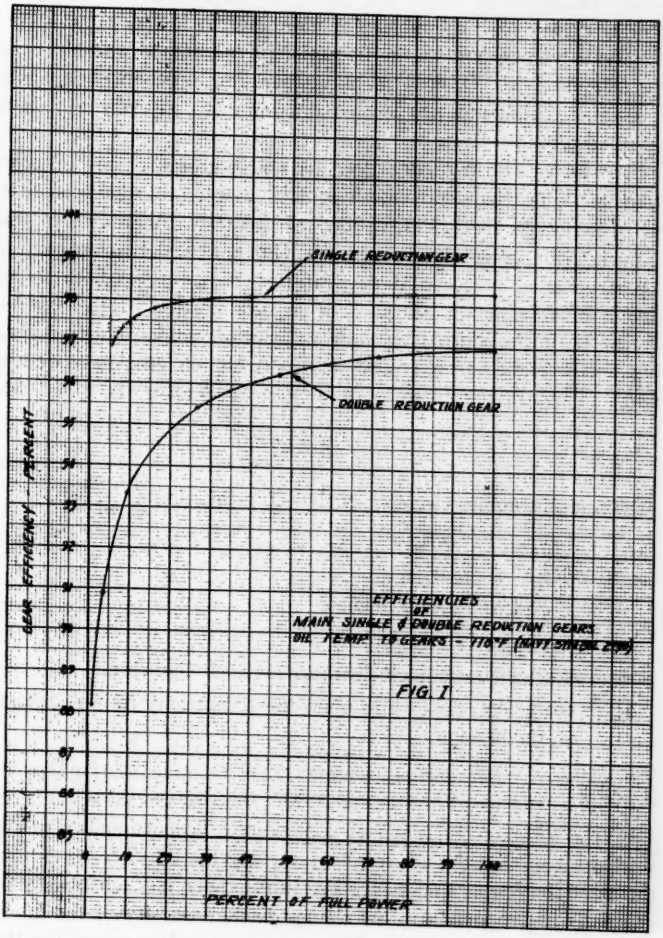

SINGLE REDUCTION GEAR
DOUBLE REDUCTION GEAR
EFFICIENCIES OF MAIN SINGLE & DOUBLE REDUCTION GEARS.
OIL TEMP. TO GEARS = 118°F (MAY SINGLE 2009)
GEAR EFFICIENCY PERCENT
PERCENT OF FULL POWER

FIG. 1

part of the gain in turbine and propeller efficiencies at low powers, due to the employment of high speed turbines and relatively slow speed propellers is, therefore, sacrificed because of the relatively low efficiency of the double reduction gear at low powers.

About 95 per cent of the operation of a surface warship is at cruising speeds. Therefore, any operating procedure that will raise the efficiency of the main reduction gears at cruising speeds is worthy of attention.

The losses in a reduction gear are chiefly due to: (*a*) oil friction in the bearings, (*b*) tooth contact losses, (*c*) rotational loss. Of these three, the first is by far the greatest. According to test reports*, the power loss at the teeth of a helical marine gear is about 0.4 of 1 per cent of the transmitted power for both single and double reduction gears. The greater portion of the loss is, therefore, in the journal bearings, of which a single reduction gear has six or eight and a typical double reduction gear has twenty-two. The fraction torque in a journal bearing is given by,

$$T = K_f \mu N R^2 \frac{L}{m}$$

where, μ = absolute viscosity of the oil in the film, $\frac{\text{lbs. sec.}}{\text{sq. inch}}$

T = torque, lb.-inches.

N = RPM. of journal.

R = radius of journal, inches.

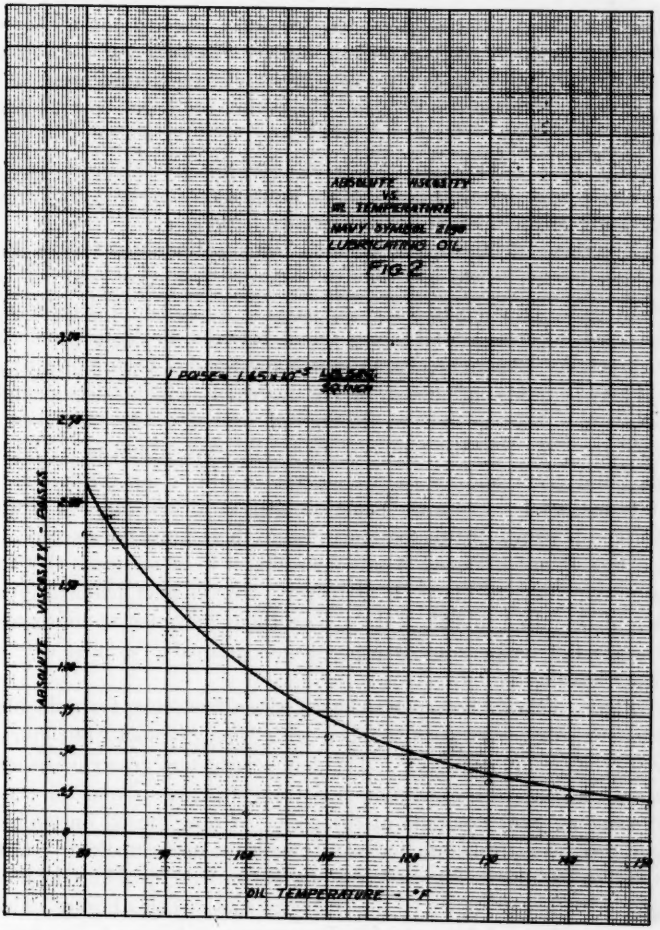
L = bearings, length, inches.

m = clearance ratio, inch clearance per inch diameter.

K_f = coefficient, depending upon the eccentricity of the journal in the bearing.

The value of "K_f" decreases somewhat with increase of viscosity, "μ," when the speed of rotation, "N," remains the same. Therefore, the torque "T" deviates from direct proportionality to the viscosity "μ," but the net result of varying the oil viscosity is that the friction torque, and hence the horsepower loss, changes substantially in proportion to the change in the viscosity of the lubricating oil in the bearings. This statement is less exact for the high speed pinion bearings than for the lower speed bearings, but

* "Mechanical Losses for Large Power Transmission," by L. M. Douglas in North East Coast Institute of Engineers and Shipbuilders, Dec. 1940.

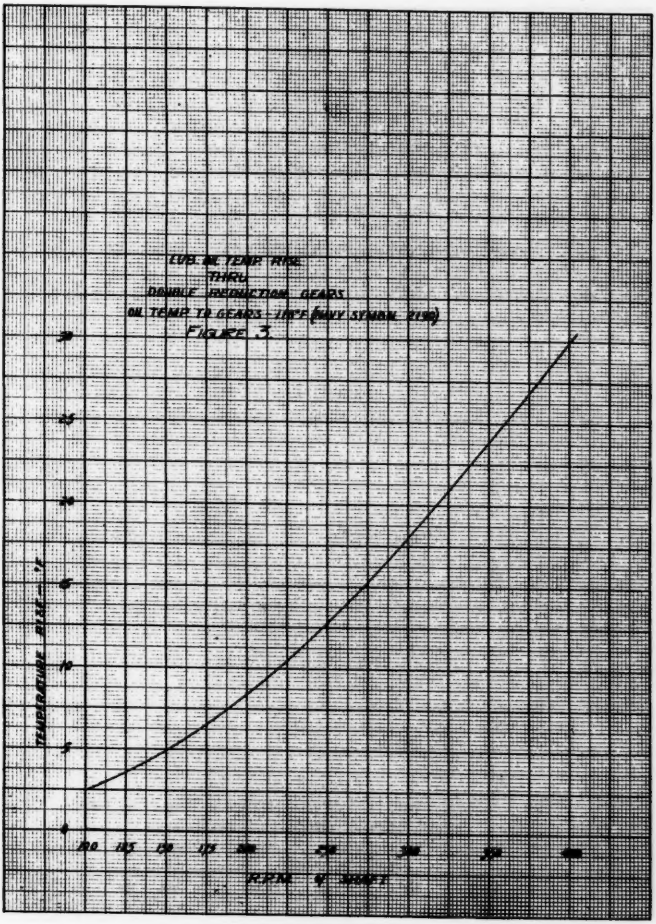

ABSOLUTE VISCOSITY
VS
OIL TEMPERATURE
NAVY STANDARD 2190
LUBRICATING OIL.
FIG 2
1 POISE = 1.45 x 10⁻⁵ LBS SEC/SQ INCH
ABSOLUTE VISCOSITY - POISES
OIL TEMPERATURE - °F

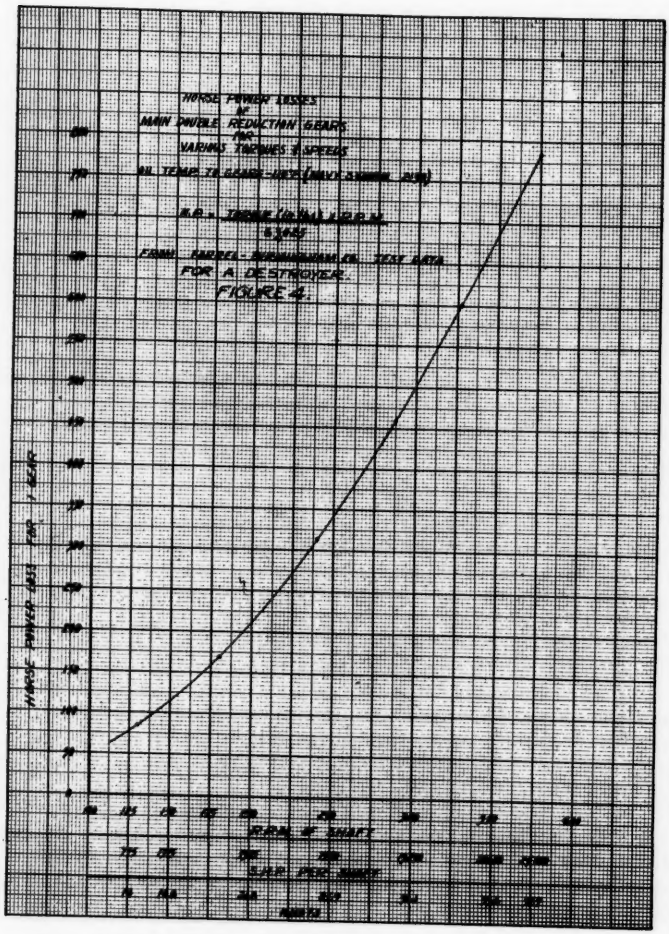
represents average variation of losses with change in oil viscosity for all of the reduction gear bearings.

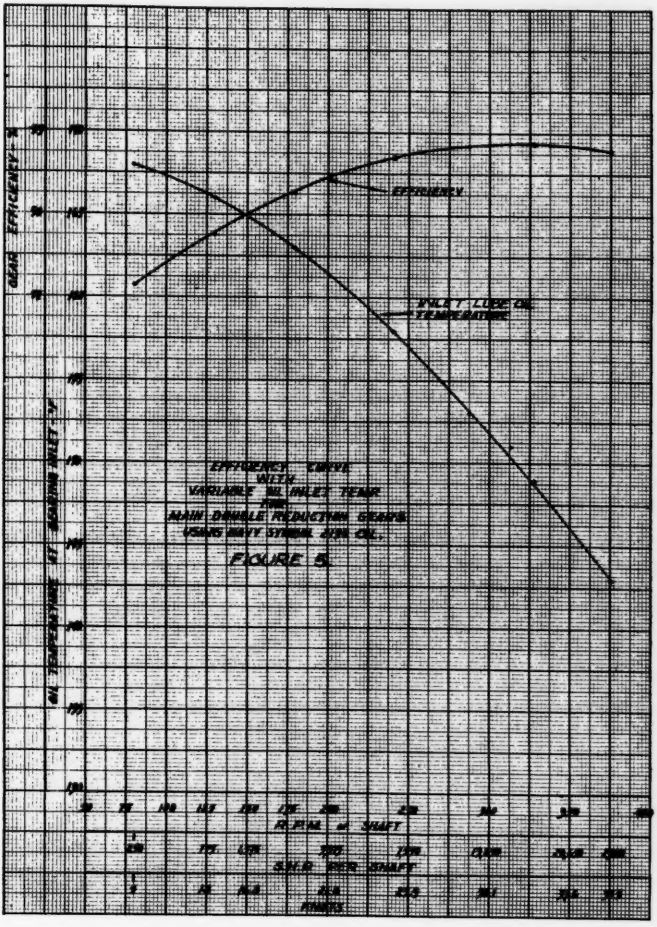
Figure 2 shows the variation of absolute viscosity of Navy Symbol 2190 lubricating oil with temperature. Raising the oil temperature from 90 degrees to 110 degrees F. changes the absolute viscosity 50 per cent, but an increase in oil temperature from 130 degrees to 150 degrees F., changes the viscosity only about 34 per cent. Horsepower losses in the bearings will change in approximately direct proportion. It follows that the horsepower loss in the gear may be controlled within limits by controlling the temperature of the inlet oil to the bearings of the gear.

The curve of Figure 3 shows the average temperature rise of the lubricating oil in the bearings of a double reduction gear for a destroyer. At cruising speeds a rise of 3 degrees to 5 degrees F. is noted, whereas at full power this is between 25 degrees and 30 degrees F. Experience shows that Navy Symbol 2190 lubricating oil may be used continuously at 170 degrees F. without detrimental effect to either the teeth or the journal bearings. At full power, therefore, the oil inlet temperature to the gears could be as high as 140 degrees, and the oulet temperature still be not greater than 170 degrees F. Since the same oil is used to lubricate the main turbine bearings, the temperature rise in these must not exceed 30 degrees F. at full power if safe operation is to be attained. Up to approximately 16 knots in destroyers, the oil inlet temperatures might be raised to 165 degrees F. without detrimental effect, although this temperature is probably not attainable without externally heating the oil.

Figure 4 shows horsepower losses, taken from test, for a double reduction destroyer gear supplied with Navy Symbol 2190 lubricating oil at 110 degrees F. The double reduction gear efficiencies shown in Figure 1 were computed from these losses and the corresponding SHP. outputs of the gear. If now the oil inlet temperature be controlled in accordance with the curve marked " Inlet Oil Temperature," Figure 5, the efficiency of the double reduction gear will vary as shown by the efficiency curve of this Figure. The curve marked " Inlet Lube Oil Temperature," Figure 5, has been so drawn that the sum of the temperature rise in the bearing and the inlet oil temperature will give a constant bearing oil temperature, namely, 170 degrees F. The variation in range of efficiency


LUBL OIL TEMP RISE
THRU
DOUBLE REDUCTION GEARS
OIL TEMP TO GEARS - 110°F (NAVY STEAM PLAN)
FIGURE 3.
TEMPERATURE RISE — °F.
R.P.M. OF MOTOR


HORSE POWER LOSSES
IN
MAIN DOUBLE REDUCTION GEARS
FOR
VARIOUS TORQUES & SPEEDS
FROM PARSEL-SHERMAN EX. TEST DATA
FOR A. DE STROBEL.
FIGURE 4.


EFFICIENCY
INLET LUBE OIL TEMPERATURE
EFFICIENCY CURVE WITH VARIABLE OIL INLET TEMP. ON MAIN DOUBLE REDUCTION GEARS USING MOTIV STEAM TURB. OIL.
FIGURE 5.

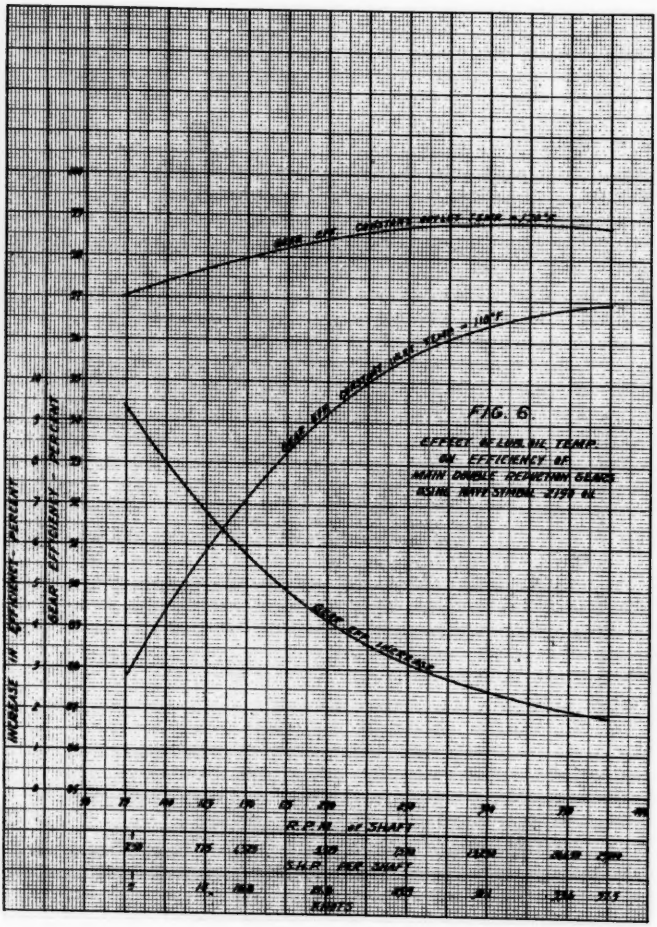


FIG. 6.

EFFECT OF LUB. OIL TEMP. ON EFFICIENCY OF MAIN DOUBLE REDUCTION GEARS. R.M.S. ARIZONA. 2150 H.P.

will then be less than 2 per cent between 15 knots and full power. Referring back to Figure 1, it will be noted that since 15 knots corresponds to about one twenty-fifth of full power, the efficiency of a double reduction gear varies from 90.9 per cent to 97 per cent over this range, when the oil inlet temperature is held constant at 110 degrees F.

In Figure 6 are plotted, (*a*) the efficiencies of a double reduction gear operating with inlet oil temperature kept constant at 110 degrees F., (*b*) the efficiencies that will result if the outlet temperature is kept constant at 170 degrees F. The resulting gain in efficiency is also shown. It will be noted that the gain is about 8.5 per cent at 9 knots and 1.9 per cent at full power.

Objection may be made to increasing lubricating oil inlet temperatures on the ground that the oil will have to be changed more often. The saving in fuel oil however, will be far greater than the added cost of lubricating oil.

It will probably not be possible to raise oil inlet temperatures as high as suggested for the lower powers, since equilibrium between heat generated and heat liberated may be established before 165 degrees F. temperature is reached. This naturally suggests heating the lubricating oil at lower powers and cooling it at the higher powers, the temperature being automatically controlled by a thermostat. However, it is felt that some gain at low power operation is possible by shutting off the lubricating oil cooling water and allowing the oil temperature to approach values indicated herein.

The writer gratefully acknowledges his indebtedness to Prof. G. B. Karelitz, of Columbia University, for his helpful suggestions and criticisms of this paper.

DISCUSSION.

By Lt. Commander W. C. Latrobe, U.S.N., and
Lt. (jg) J. C. Radford, U.S.N.R.

Mr. R. Michel's paper, "Efficiency of Double Reduction Gears as Influenced by Lubricating Oil Temperatures," has been read with considerable interest. It is desired, however, that certain of the probable effects of increased temperatures upon the lubricating oil be emphasized.

The remarkable development in the quality of lubricating oils over the past few years has been paralleled by an increase in the requirements for these materials. At any given stage in this development, it may safely be said that designers, manufacturers and users of equipment requiring lubrication have consistently stepped up lubricating requirements beyond the abilities of the oil at the time. The production of improved lubricants by the refiner has contributed, together with the use of alloy materials, higher temperatures and pressures, and greater rotative speeds, toward a greatly reduced weight/power factor and increased efficiency of operation. This applies to the internal combustion engine as well as the steam turbine.

Turbine oils in general use at the present time are exceedingly well refined petroleum oils; however, the fact remains that they are hydrocarbon materials and, as such, will oxidize and decompose at elevated temperatures. This process of oxidation and decomposition is dependent upon several factors, namely, temperature, time, pressure, oxygen, and occurrence of catalytic agents, all of which, with the possible exception of pressure, are present in turbine lubrication.

The oxidation of lubricating oil is accompanied by the formation of sludge and organic acids resulting, in turbine oils, in decreased surface tension and oil/water interfacial tension. This condition leads to increased emulsion formation and air entrainment, with consequent operational difficulties.

The occurrence of oxygen in a turbine lubricating system results from the presence of entrained air, the reaction with the turbine oil in a given system being dependent upon the temperature and the time interval involved. Laboratory tests have indicated that the critical temperature of this oxidation is approximately 140 degrees-150 degrees F.; above this point, the rate of oxidation being doubled for each 20 degrees F. temperature rise. Inasmuch as the useful life of a turbine oil is dependent upon the deterioration which has occurred, this substantially means that in this range, an increase of 20 degrees F. in operating temperature will reduce by one-half the effective life of the lubricating oil.

The reaction of oxygen with a hydrocarbon material is greatly influenced by the presence of catalytic materials. Iron, copper and water, which are present in turbine systems, fall into this classification.

In addition, it is desired to point out that the rate of rusting of the steel portions of a turbine lubricating system is increased with increased temperatures. This phenomenon is not a result of oil deterioration, but, rather, of the inability of highly refined mineral oils to " wet " the steel surfaces, the result being an oxygen and water corrosion and consequent danger of clogging of oil passages and of scoring of bearings by the particles of rust.

When normal maintenance procedure is followed, rusting is not a large factor, but when operation will not permit overhaul at the least variation from optimum conditions, as the occurrence of fresh and salt-water leakage into the system, rusting may become serious.

Mr. Michel has shown that the friction torque in a journal bearing is a function of the absolute viscosity of the oil in the film and the speed of the journal. It is noted, however, that the load carrying capacity of the bearing has a similar relation to these factors. It follows, therefore, that if too low a viscosity obtains, in the event of sub-standard bearing alignment, failure may occur at low speeds under shock loading.

The application of Symbol 2190 oil to the turbine systems of Naval vessels has resulted from a compromise between lubrication of the journal bearings and lubrication of the reduction gear teeth. Whereas 2190 is slightly heavy for the former, the use of a lighter oil for reduction gears has been definitely frowned upon by gear manufacturers. It appears, therefore, that if a lower viscosity

bearing lubricant be deemed necessary, provision must be made to supply a heavier material to the gear teeth to prevent undue wear.

Three methods of providing lighter bearing lubricating oil appear possible, namely:

(*a*) Increasing the bearing outlet temperature to a predetermined optimum value, which may best be accomplished by controlling the oil cooler and oil circulation. Because of the danger of overheating the oil, under no circumstances should heat be deliberately applied to the lubricating oil from an external source. It will probably be necessary to supply means for cooling the oil to the gear teeth, as the rate of circulation in the present day marine installation is too great to allow much cooling and settling action.

(*b*) The use of a lighter grade of oil, say Symbol 2135, at lower temperatures. Adequate cooling must be given the oil to the gears in order that sufficient lubrication will obtain. Assuming the application of a typical 2190 oil to the reduction gear teeth at a temperature of 150 degrees F., in order to obtain the same viscosity with a typical 2135 oil it will be necessary to operate at 135 degrees F., at which temperature comparable lubrication will result. It is noted that the temperature rise of 2135 oil in a journal bearing due to internal friction will be less than that observed with 2190 oil and that by increased fluidity, greater circulation and cooling action will result. Stabilized temperatures of these two oils recorded during Navy Work Factor determinations were 218 degrees F. and 238 degrees F., respectively—a 20 degree F. difference.

(*c*) The adoption of separate lubrication systems for bearings and gears, enabling the use of a lighter grade of oil in the former and a heavier or, better yet, an extreme pressure oil, in the latter. The use of E. P. (extreme pressure) lubricants in gears will allow design and use of more compact units with higher tooth pressures.

UNUSUAL FEATURES OF FABRICATION AND HULL ERECTION OF 165-FOOT ALL WELDED SUBMARINE CHASERS.

By Lt. Robert Hughes, U. S. N. R.

The Dravo Corporation, Neville Island, Pittsburgh, Pa., has for many years successfully built barges and river towboats by the use of assembly line methods. Preassemblies are fabricated by use of position-weld jigs, moved to the assembly line, and, progressively, to the launching ways. (See Figure 1).

Experience indicates that, due mainly to the elimination of the fatigue factor, a better weld will be produced by the workman and welding time is reduced from 25 to 30 per cent by the use of position welding. Examination of the test records of welders submitted for the Navy test disclosed that the number of acceptances on the downhand test to the overhead was in the ratio of approximately four to one. It was thus possible, not only to reduce fabricating time, but also to use welders having average qualifications, i.e., qualified by Navy test for downhand position welding only, for much of the work, with a consequent expansion of the available labor market. With the present incentive for speed, coupled with the assembly line experience of the contractor, it was decided that similar methods of construction, with modifications where necessary, would be adopted for fabrication of the all-welded hulls of the submarine chasers.

Buckling difficulties of course were expected, particularly in the bulkheads as most of these are of 5.7 pound plate with 4½-inch × 2.38-inch × 3.75 pound Tee stiffeners at 21-inch spacing. However, it was felt that buckling could be controlled more readily if the bulkheads were fabricated prior to installation in the ship. The procedure followed was to first weld the plates together, the welds then being stretched with a flatter and sledge to remove buckles. Following this the unit was tacked to a flat steel floor prior to the application of the stiffeners. A half inch bow was put in the stiffen-

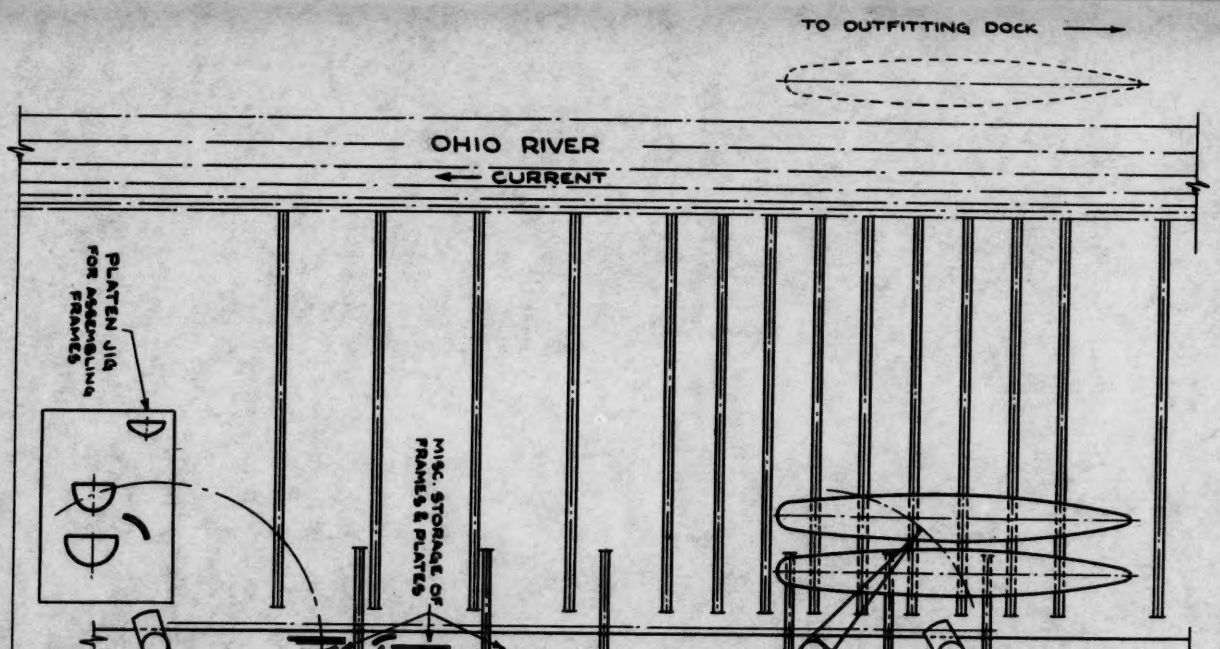
ers so that when they were forced down to the plate and tacked, the contact edges were in compression, thus offsetting the normal bowing from the subsequent welding. Welding of the stiffener to the plate was then completed and the few resultant buckles removed by the spot heat and quench method.

An interesting development was a joiner bulkhead, the stiffeners of which were formed by doubling the plate back on itself. The joiner bulkheads are of Number 16 gauge galvanized steel sheet. The panels were cut oversize and the stiffening pressed in at twelve-inch centers on a press-brake machine equipped with a special forming die. A mold of 2-¾-pound × Number 12 gauge plate was crimped around the edge and tack welded. The double edge of the stiffener was also tack welded at eighteen-inch centers to eliminate murmur. Dolphinite number 2016 Joint Glazing Compound was then forced in between the faying surfaces of the bent plate stiffeners and between the faying surfaces of the formed mold and bulkhead panel in order to make them vermin-proof, sandpapered smooth and painted in accordance with standard practice. (see Figure 2).

The hulls were built in five sections; preassemblies numbers 1, 2, 3 and 4 being built upside down to facilitate erection and fitting and in order to obtain the maximum of welding in the downhand position. Preassembly number 5 was built right side up. (see Figure 3). Shoring which would have been required to support engine girders would more than have offset the economy of the upside down construction.

Assembly jigs were erected for bow, stern and midsections upon which the deck plating was fitted and securely tacked, the jigs being laid out to the correct sheer and camber of the deck. The jigs consisted of a rigidly braced structural steel framework, resting on concrete piers making practically a permanent structure, yet one which is easily dismantled. (see Figure 4).

Transverse frames, after assembly on a platen jig located nearby, were erected on the deck, faired to wood templates and tied together with the sheer strake, longitudinal stringers and the vertical keel. The preassembled strength bulkheads were fitted into place and tacked as the work progressed. The shell plating was then fitted and tacked to the stringers and the finish welding of stringers to bulkheads and stringers to shell plating was then ac-


TO OUTFITTING DOCK
OHIO RIVER
CURRENT
PLATEN JIG FOR ASSEMBLING FRAMES
MISC. STORAGE OF FRAMES & PLATES

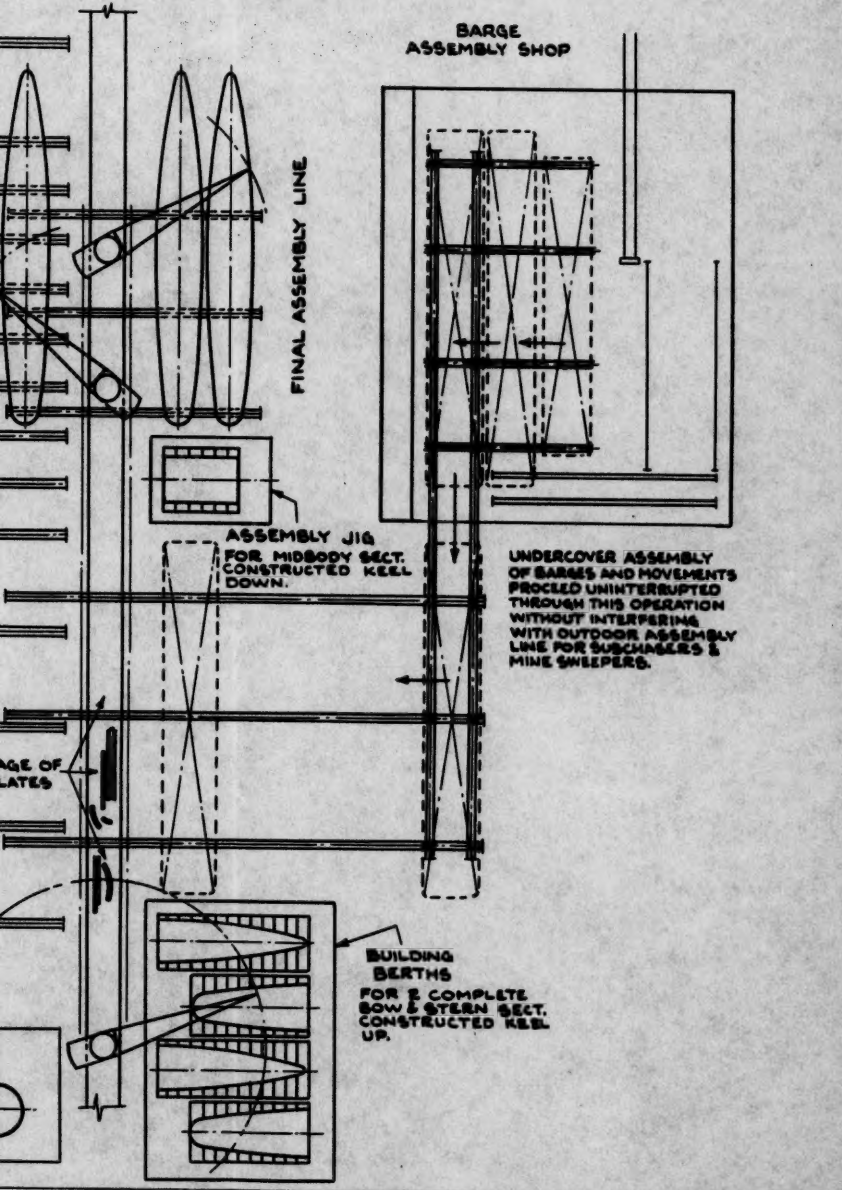
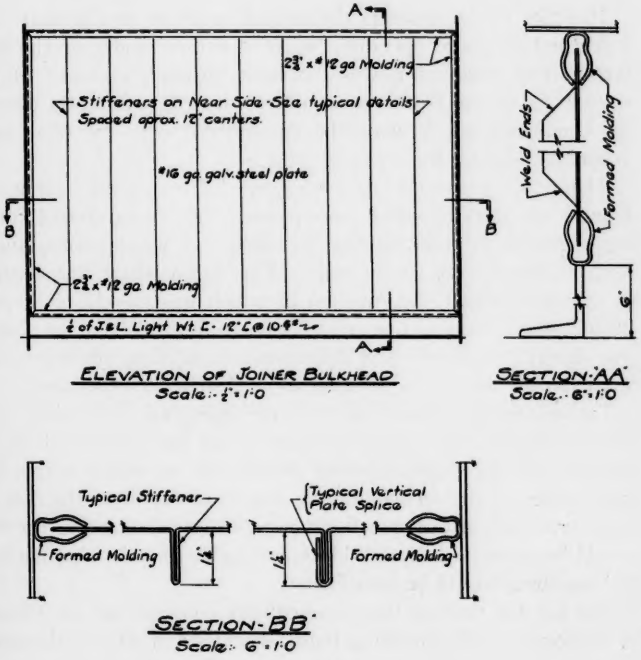

BARGE
ASSEMBLY SHOP
FINAL ASSEMBLY LINE
ASSEMBLY JIG
FOR MIDBODY SECT.
CONSTRUCTED KEEL
DOWN.
UNDERCOVER ASSEMBLY
OF BARGES AND MOVEMENTS
PROCEED UNINTERRUPTED
THROUGH THIS OPERATION
WITHOUT INTERFERING
WITH OUTDOOR ASSEMBLY
LINE FOR SUBCHASERS &
MINE SWEEPERS.
BUILDING
BERTHS
FOR 2 COMPLETE
BOW & STERN SECT.
CONSTRUCTED KEEL
UP.

FIGURE 1.



NOTES

All welding in way of Toilets, Showers and Wash Rooms to be ⅛" Cont. both sides elsewhere 6" x 2" x 4" both sides. Toe of molding to sheeting to have 6" x 1" x 4" both sides.

FIGURE 2.

complished in order to hold the shape of the vessel. The transverse frames were next welded to the plating followed by the bulkheads being welded into place. The deck butts were now welded and finally the shell laps and shell butts were completed.

Moving the preassemblies to the final erection site presented no problem as the weights were nominal and ample crane service was available. However, it was realized that critical positions would be encountered during the turning-over process, due to the rapidly moving center of gravity of the irregular sections about the center of rotation. This was particularly true of the bow section.

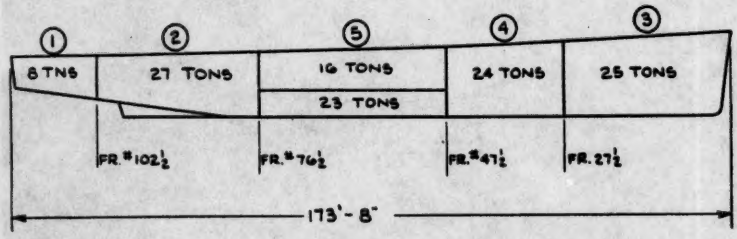
In order to demonstrate the critical positions to the boat yard force and to crane operators, a cross-section model of the bow section was made as it was the most unstable, and one of the section aft of the bow because it represented, sufficiently closely, the conditions which would be encountered with the other pre-assemblies during the turning process.

The cross-section models were shaped from ¼-inch ply-wood, around the edges of which were pasted strips of cardboard forming a groove to accommodate the sling. A wood sheave and a sling of twine were cut to scale. The sheave shaft was secured to a wooden board, the function of which was merely to prevent the ply-wood cross-section model from tilting. When the sheave was slowly turned by hand the unstable conditions to be encountered during the later operation were clearly demonstrated.

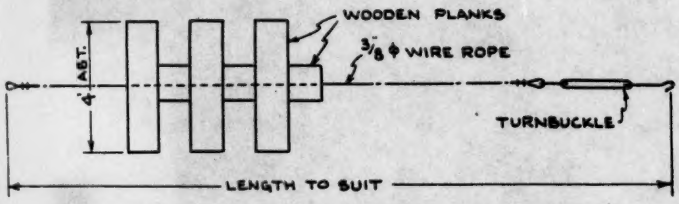
Two extremely critical points were indicated, particularly for the bow section; one when the section had been turned about 30 degrees and the other at about 60 degrees, at which points the mass center of the section was moving most rapidly. Because of these predicted critical positions, it was decided that the turning should be accomplished with block and tackle in order that hauling and snubbing would be coordinated.

The rig for turning the preassemblies consisted of an 18-inch × 70-pound C. B. equalizing beam, shackles and 30-inch diameter sheaves. One inch diameter wire rope slings were fitted around the hull, and passed over the sheaves. A belt of wooden planks, three frame spaces wide, held together by means of a ⅜-inch diameter wire rope strung through them, then tightened around the hull by means of a turnbuckle, was placed between the slings and the hull for protection. A gunwale protecting timber was also fitted in way of the turning cables, and the hull shored internally to prevent squeezing. (see Figure 3).

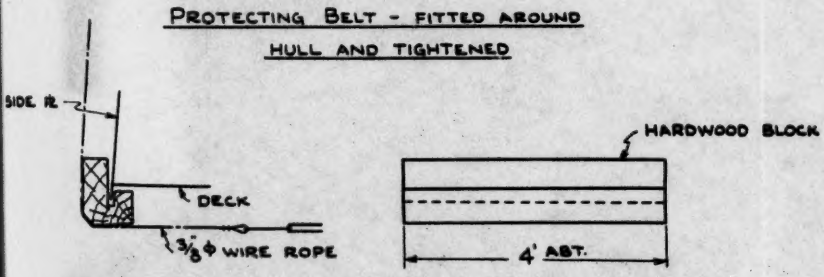
A preassembly was lifted and brought to the final erection site, and, while the main hooks of two gantry cranes held it in the air, the turning was accomplished by the block and tackle, one end of the line being secured to a lug welded to the hull. As a measure of precaution, and in order to overcome the tendency of the preassembly to roll back at the critical positions, another turning line was connected to the whip hook of one crane. Similarly snubbing was done through a block and tackle and again a separate snubbing



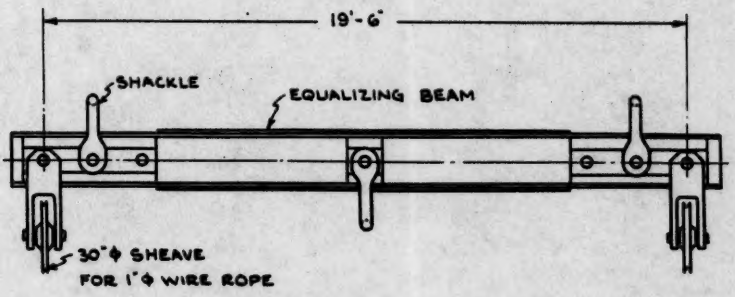
HANDLING WEIGHTS FOR FIELD PREASSEMBLIES



PROTECTING BELT - FITTED AROUND HULL AND TIGHTENED



PROTECTING TIMBER AT GUNWALE



TURNING DEVICE

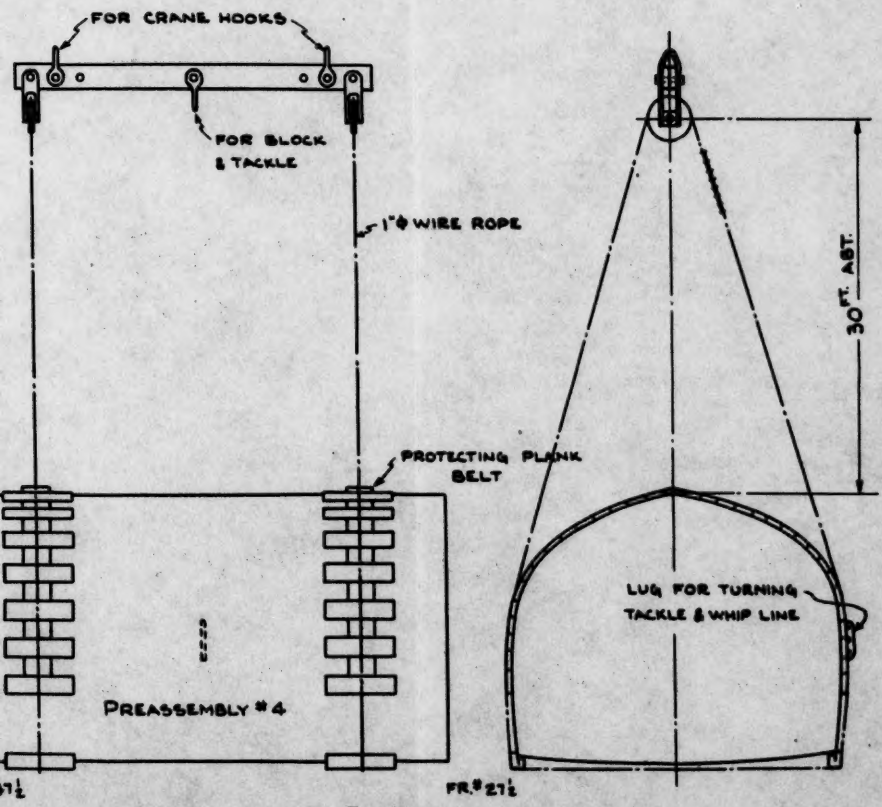
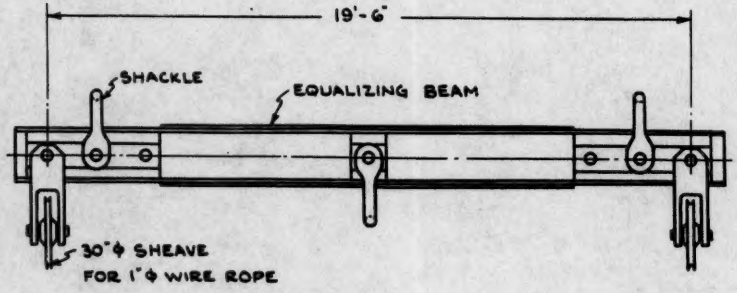
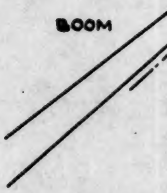


FIGURE 3.





TURNING DEVICE

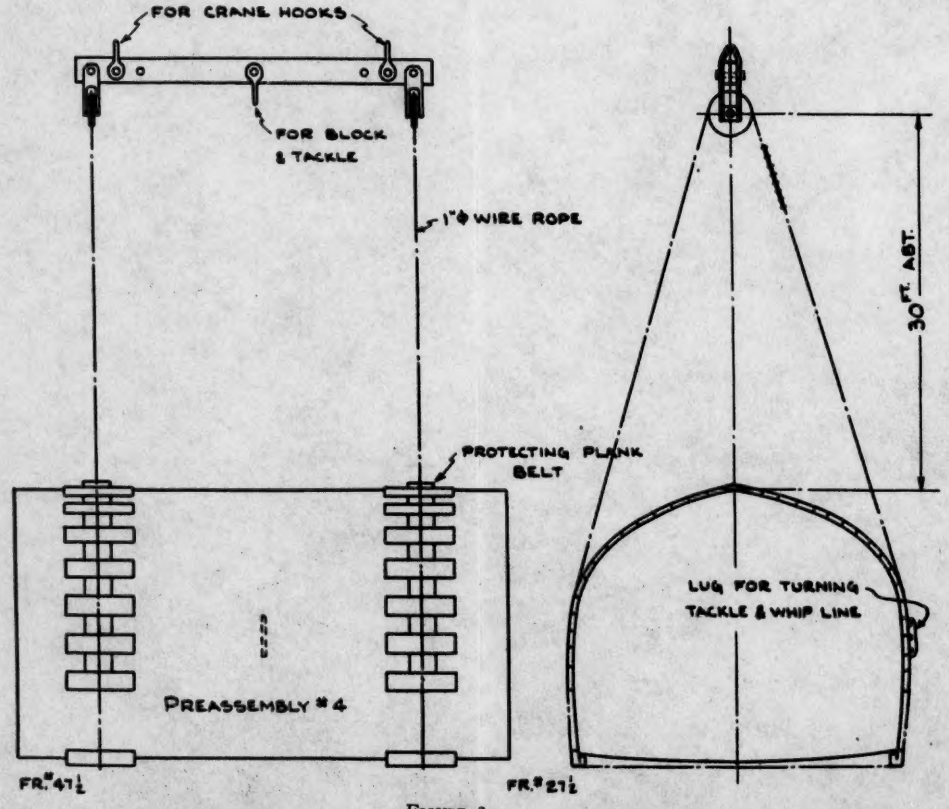
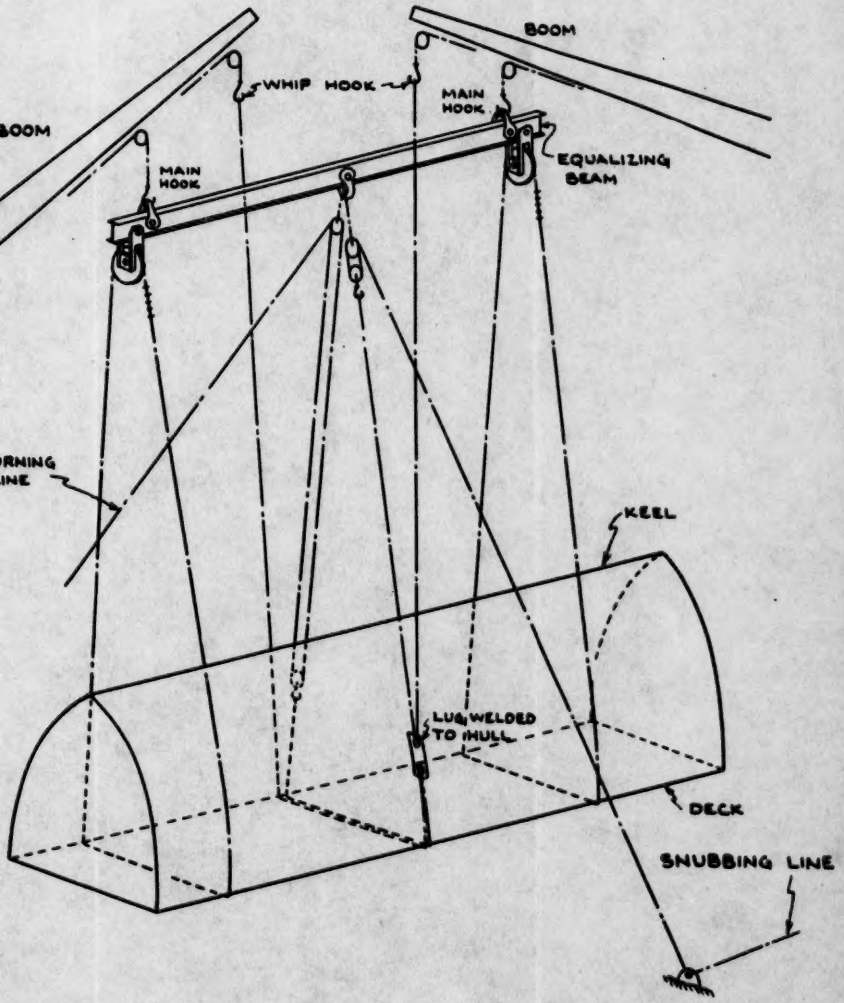


FIGURE 3.


BOOM
WHIP HOOK
MAIN HOOK
BOOM
MAIN HOOK
EQUALIZING BEAM
KEEL
LUG WELDED TO HULL
DECK
SNUBBING LINE

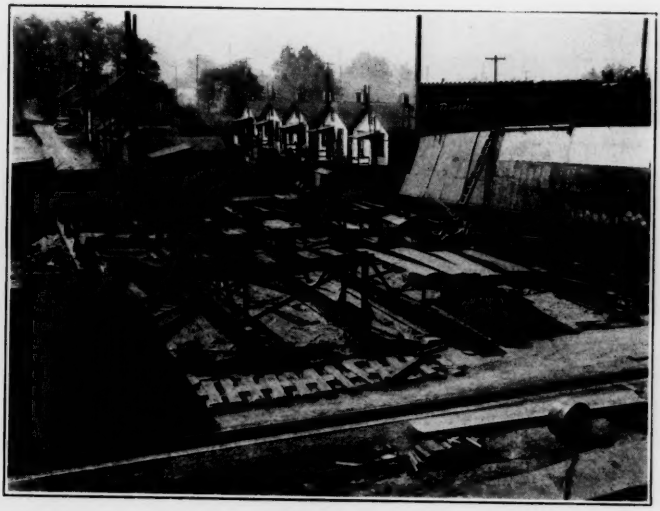


FIGURE 4.—ASSEMBLY JIGS FOR BOW AND STERN PREASSEMBLIES.

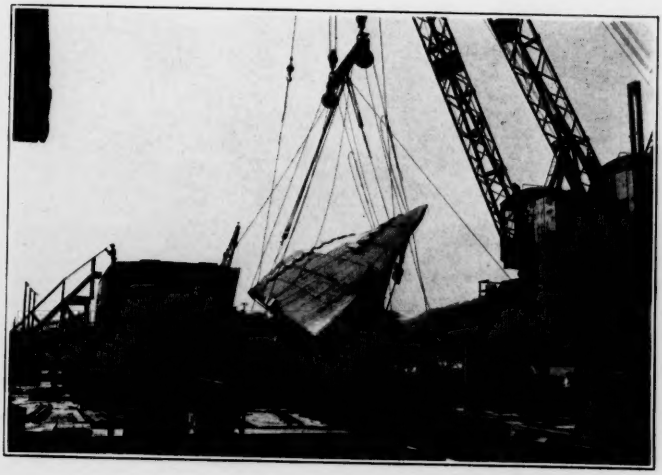


FIGURE 5.—TURNING THE BOW PREASSEMBLY.

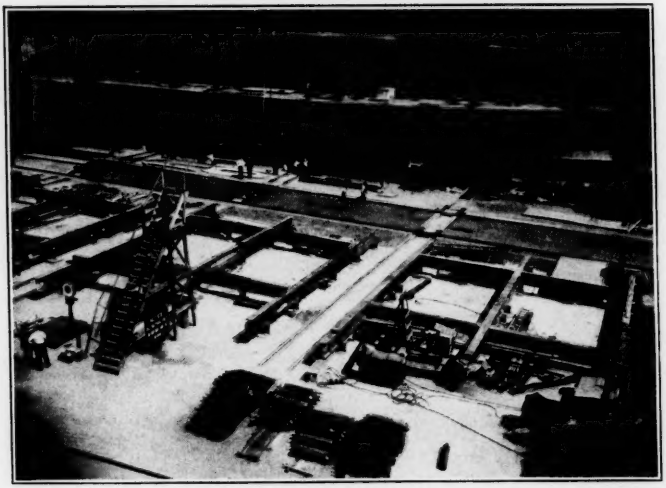


FIGURE 6.—GENERAL VIEW OF ASSEMBLY WAYS AFTER FIRST HULL WAS MOVED TO LAUNCHING WAYS.

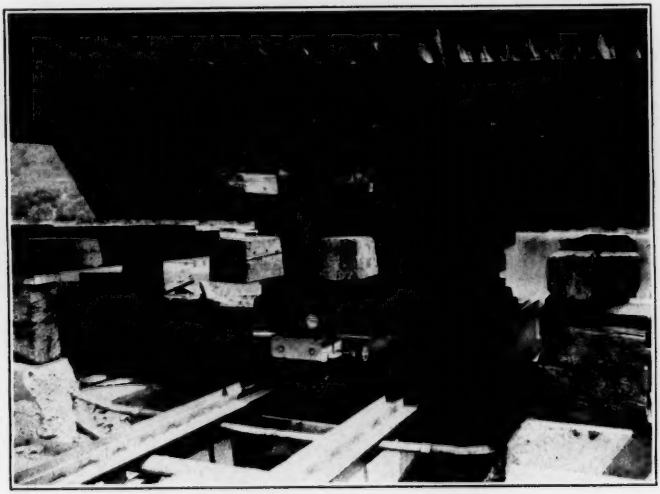


FIGURE 7.—VIEW OF TRANSFER CARRIAGE—HULL LIFTED AND READY TO BE MOVED TO LAUNCHING WAYS.

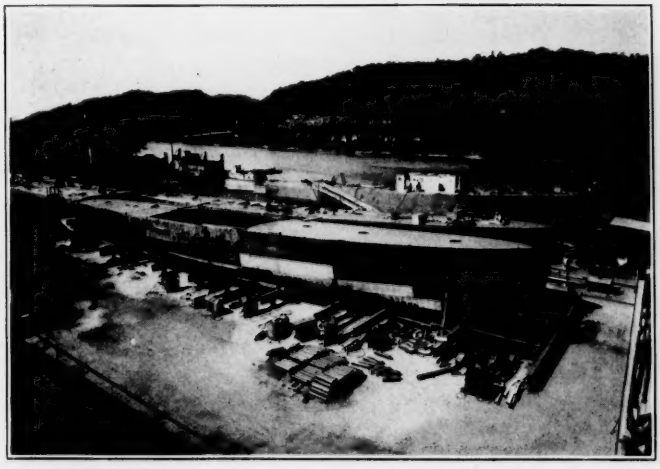


FIGURE 8.—ASSEMBLY WAYS IN FULL OPERATION—HULLS MOVE PROGRESSIVELY TO LAUNCHING WAYS.

line was attached to the whip hook of the second crane. The turning of the preassembly was thus under control at all times. (see Figure 5).

Assembly ways, made up of 24-inch I-beams, capable of holding two complete hulls had been laid (see Figure 6), and the midsection, comprising the main and auxiliary engine rooms which as previously stated was built right side up as a preassembly in a separate location, was chocked in place.

Upon completion of the turning operations of the forward and after sections they were set in place, and, while still held by the main hooks of the two gantry cranes, pulled home by means of turnbuckles, which were secured to lugs welded to each section. About 18 inches of the lap of the shell plates had been left unwelded to facilitate fitting and to assist in controlling the over-all length of the vessel at this time. Care having been exercised in laying down the assembly ways, no difficulty was experienced in fairing the hull during final assembly.

It will be noted (Figure 6) that the assembly ways were built in four sections, and between each section grooved tracks laid. The tracks were made of 6-inch channels secured, flanges up, to 24-inch steel beams supported on concrete foundations. Upon completion of the hull welding the entire structure was moved to the launching ways by means of five transfer carriages, which were hauled along the tracks by winches. The transfer carriages were distributed under the hull as follows:—two approximately half way between the bow and midships (one port and one starboard), two midships (port and starboard), and one under the skeg approximately half way between midships and stern.

Each transfer carriage is an all-welded framework carried on four 18-inch diameter, 4½-inch width, solid wheels cut from hot rolled steel billets. The two lifting beams of the carriage to which cushioning timbers are attached, are raised by four 15-ton hydraulic jacks actuated from one pump in order to obtain equalization of load. The load between the five transfer carriages is then controlled within limits, by careful observance of the pressures registered on the gauges attached to each carriage. (see Figure 7).

In conclusion it appears that the benefits derived by the use of production line assembly on small vessels are (1) A substantial reduction in fitting and welding time; (2) An overall improvement

in the quality of the weld in direct ratio to the amount of position welding obtainable. (3) Building the vessel in sections allows relief points for locked up or internal stresses due to welding shrinkage and permits adjustment of proper overall length of the vessel; (4) A possibility of using a greater number of welders with average qualifications, i.e., qualified for downhand welding only; (5) Conservation of launching ways.

A SIMPLIFICATION IN THE MEYER-HOVGAARD SYSTEM OF PIPE STRESS FORMS.

By W. E. Bleick.*

Now that the details of the " Meyer-Hovgaard " method of pipe stress calculation have been published by M. S. Noyes †, it is thought worthwhile to describe a modification of this method, which simplifies the calculation forms and reduces the number required by 67 per cent.

This simplification has been accomplished by making use of the well known principle that the first and second moments of arc length, i.e., the coefficients A_x, A_y, A_{x2}, A_{y2}, and A_{xy}, achieve their simplest mathematical form when expressed with reference to the center of gravity of the arc. Use of this principle provides such simple formulas that the coefficients for all three coordinate planes of a pipe section may be calculated on a single form. Thus Forms No. 1 to 45 of Noyes' paper are replaced by Forms No. 302 to 316 of this paper.

Inspection of Forms No. 300 to 316 shows that L_x and L_y are the coordinates of the center of gravity of a straight pipe section, and of the center of curvature of a circular pipe section. This definition of L_x and L_y differs from that in Noyes' paper. The symbols l_x and l_y of Forms No. 300 to 316 are the coordinates of the center of gravity of the pipe section. For straight pipe sections it is obvious that $l_x = L_x$ and $l_y = L_y$.

Form No. 298 is a chart for the selection of the pipe stress forms and corresponds to Form No. 55 of Noyes' paper. If, for example, the orientation of a circular pipe section corresponds to the position 7b of Form No. 298, then the XY-plane, YZ-plane and ZX-plane coefficients are calculated in columns (b), (c) and (a) respectively, of Form No. 307.

* Civilian Instructor, Dept. of Mathematics, U. S. Naval Academy.
† This Journal, vol. 53, No. 2, May 1941.

Form No. 300 is a basic form which may be used for straight pipe sections in any position in a plane parallel to a coordinate plane. Form No. 301 is a basic form which may be used for circular pipe sections of any angular size in any position in a plane parallel to a coordinate plane. Note that the angle Φ is expressed in radians. In using Forms No. 300 and 301 care must be taken to attach the proper algebraic sign to the sine and cosine functions, depending upon the quadrant in which the angles θ and Φ lie.

The author is indebted to Macon Fry for checking the mathematical calculations involved in constructing the forms of this paper.

Comment by the author (MR. MASON S. NOYES).—Mr. Bleick's simplificaton of the Meyer-Hovgaard method for determining thermal expansion stresses in piping is welcomed as a definite addition to the method and the writer expresses his appreciation for its submittal.

Time has not permitted more than a quick check-up of the validity of the formulae. The writer wishes to reserve opinion as to the full utility of Mr. Bleick's simplification until usage has demonstrated whether or not additional time and complication is involved in determining the extra set of dimensions for the centers of each element in contradistinction to the considerable simplification of the formulae for the "A" coefficients.

SUBJECT:

CHART FOR THE SELECTION OF PIPE STRESS FORMS.

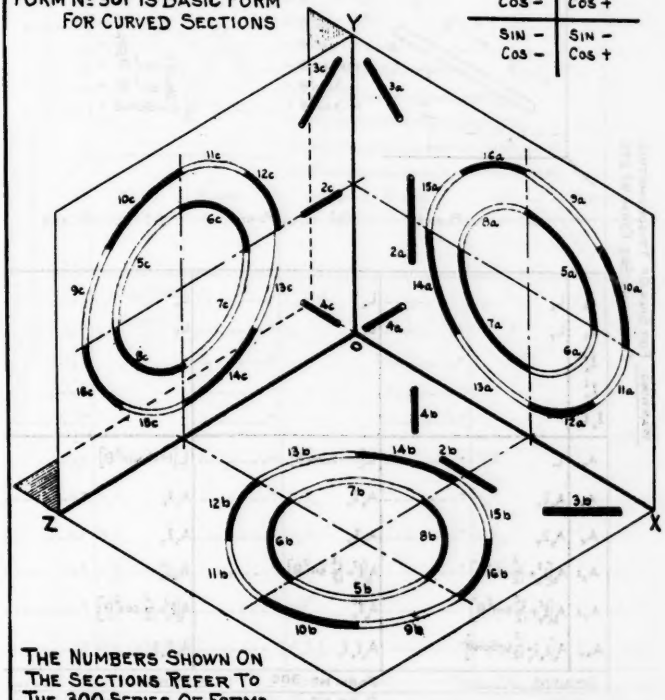
COMPUTED BY

CHECKED BY

DATE:

FORM Nº 300 IS BASIC FORM
FOR STRAIGHT SECTIONS
FORM Nº 301 IS BASIC FORM
FOR CURVED SECTIONS

SIN + COS −	SIN + COS +
SIN − COS −	SIN − COS +



THE NUMBERS SHOWN ON
THE SECTIONS REFER TO
THE 300 SERIES OF FORMS

GROUP Nº

FORM No. 298

SHEET Nº

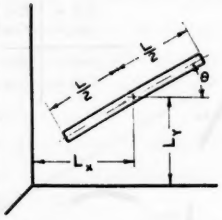
SHEET OF

SUBJECT: COEFFICIENTS OF ___ PIPE

COMPUTED BY. _____ DATE _____

CHECKED BY _____

SECTION Nº ___



θ = _____

$\cos^2\theta$ = _____

$\sin^2\theta$ = _____

$\cos\theta\sin\theta$ = _____

$1+3\cos^2\theta$ = _____

$1+3\sin^2\theta$ = _____

L = _____

$\frac{L^2}{12}$ = _____

$\frac{L^2}{12}\cos^2\theta$ = _____

$\frac{L^2}{12}\sin^2\theta$ = _____

$\frac{L^2}{12}\sin\theta\cos\theta$ = _____

(a) ___ PLANE (b) ___ PLANE (c) ___ PLANE

WARNING: FOR SIGN OF TRIGONOMETRIC FUNCTIONS SEE FORM Nº 298

l_x	L_x = _____	L_x = _____	L_x = _____
l_y	L_y = _____	L_y = _____	L_y = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	L = _____	$L[1+3\cos^2\theta]$ = _____	$L[1+3\sin^2\theta]$ = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A_s\left[l_x^2+\frac{l^2}{12}\cos^2\theta\right]$ = _____	$A_s\left[l_x^2+\frac{l^2}{12}\sin^2\theta\right]$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A_s\left[l_y^2+\frac{l^2}{12}\sin^2\theta\right]$ = _____	$A_s l_y^2$ = _____	$A_s\left[l_y^2+\frac{l^2}{12}\cos^2\theta\right]$ = _____
A_{xy}	$A_s\left[l_x l_y+\frac{l^2}{12}\sin\theta\cos\theta\right]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

GROUP Nº ___ FORM Nº 300 SHEET Nº _____

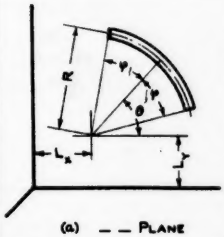
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS OF PIPE

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ____



$A = \frac{R^2}{2}\left[1+\frac{3\sin^2\varphi}{2\varphi}\right]-\left[\frac{R\sin\varphi}{\varphi}\right]^2$ = _____ K = _____

$B = \frac{R^2}{2}\left[1-\frac{3\sin^2\varphi}{2\varphi}\right]$ = _____ φ = _____ θ = _____

$C = 2.3\varphi - .18\sin 2\varphi\cos 2\theta$ = _____ R = _____

$D = 2.3\varphi + .15\sin 2\varphi\cos 2\theta$ = _____ R^2 = _____

$E = R[2.45\sin\varphi\sin\theta - .05\sin 3\varphi\sin 3\theta]/C$ = _____

$F = R[2.45\sin\varphi\cos\theta + .05\sin 3\varphi\cos 3\theta]/D$ = _____

$G = R^2[1.225\varphi - .65\sin 2\varphi\cos 2\theta + .01875\sin 4\varphi\cos 4\theta]$ = _____

$H = R^2[1.225\varphi + .65\sin 2\varphi\cos 2\theta + .01875\sin 4\varphi\cos 4\theta]$ = _____

WARNING: FOR SIGN OF TRIGONOMETRIC FUNCTIONS SEE FORM Nº 298

	(a) ___ PLANE	(b) ___ PLANE	(c) ___ PLANE
L_x	-------	-------	-------
L_y	-------	-------	-------
l_x	$L_x + \frac{R\sin\varphi\cos\theta}{\varphi}$ = _____	$L_x + E$ = _____	L_x = _____
l_y	$L_y + \frac{R\sin\varphi\sin\theta}{\varphi}$ = _____	L_y = _____	$L_y + F$ = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	$2KR\varphi$ = _____	RC = _____	RD = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A\left[\frac{l_x^2}{2}+\frac{A\cos 2\theta+B\cos 4\theta}{2}\right]$ = _____	$A[l_x^2 - E^2]+G$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A\left[\frac{l_y^2}{2}+\frac{A\cos 2\theta+B\cos 4\theta}{2}\right]$ = _____	$A_s l_y^2$ = _____	$A[l_y^2 - F^2]+H$ = _____
A_{xy}	$A\left[l_x l_y(A-B)\frac{\sin 2\theta}{2}\right]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

GROUP Nº ___ FORM Nº 301 SHEET Nº _____

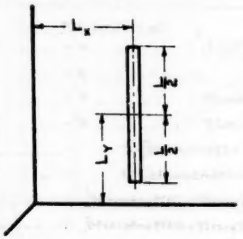
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS OF ___ PIPE ___

COMPUTED BY ___ DATE ___

CHECKED BY ___

SECTION Nº ___



L = ___
$1{\cdot}3L$ = ___
$\frac{L^2}{12}$ = ___

(a) ___ PLANE (b) ___ PLANE (c) ___ PLANE

l_x	L_x	=	L_x	=	L_x	=	
l_y	L_y	=	L_{y_1}	=	L_y	=	
l_x^2		=		=		=	
l_y^2		=		=		=	
$l_x l_y$		=		=		=	
A_s	L	=	L	=	$1{\cdot}3L$	=	
A_x	$A_s l_x$	=	$A_s l_x$	=	$A_s l_x$	=	
A_y	$A_s l_y$	=	$A_s l_y$	=	$A_s l_y$	=	
A_{x^2}	$A_s l_x^2$	=	$A_s[l_x^2 + \frac{L^2}{12}]$	=	$A_s l_x^2$	=	
A_{y^2}	$A_s[l_y^2 + \frac{L^2}{12}]$	=	$A_s l_y^2$	=	$A_s l_y^2$	=	
A_{xy}	$A_s l_x l_y$	=	$A_s l_x l_y$	=	$A_s l_x l_y$	=	

GROUP Nº ___ FORM Nº 302 SHEET Nº ___

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SUBJECT: COEFFICIENTS OF ___ PIPE ___

COMPUTED BY ___ DATE ___
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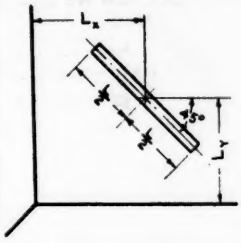
SECTION Nº ___

L = ___
$1.15L$ = ___
$\frac{L^2}{24}$ = ___



(a) ___ PLANE (b) ___ PLANE (c) ___ PLANE

l_x	L_x	=		L_x	=		L_x	=
l_y	L_y	=		L_y	=		L_y	=
l_x^2		=			=			=
l_y^2		=			=			=
$l_x l_y$		=			=			=
A_s	L	=		$1.15L$	=		$1.15L$	=
A_x	$A_s l_x$	=		$A_s l_x$	=		$A_s l_x$	=
A_y	$A_s l_y$	=		$A_s l_y$	=		$A_s l_y$	=
A_{x^2}	$A_s\left[l_x^2+\frac{L^2}{24}\right]$	=		$A_s\left[l_x^2+\frac{L^2}{24}\right]$	=		$A_s l_x^2$	=
A_{y^2}	$A_s\left[l_y^2+\frac{L^2}{24}\right]$	=		$A_s l_y^2$	=		$A_s\left[l_y^2+\frac{L^2}{24}\right]$	=
A_{xy}	$A_s\left[l_x l_y-\frac{L^2}{24}\right]$	=		$A_s l_x l_y$	=		$A_s l_x l_y$	=

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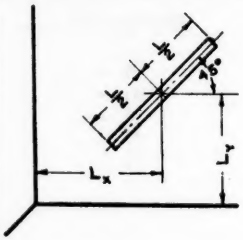
L = ---

1·15 L = ---

$\frac{l^2}{24}$ = ---



(a) ___ PLANE (b) ___ PLANE (c) ___ PLANE

l_x	L_x	=	L_x	=	L_x	=	
l_y	L_y	=	L_y	=	L_y	=	
l_x^2		=		=		=	
l_y^2		=		=		=	
$l_x l_y$		=		=		=	
A_s	L	=	1·15 L	=	1·15 L	=	
A_x	$A_s l_x$	=	$A_s l_x$	=	$A_s l_x$	=	
A_y	$A_s l_y$	=	$A_s l_y$	=	$A_s l_y$	=	
A_{x^2}	$A_s\left[l_x^2+\frac{l^2}{24}\right]$	=	$A_s\left[l_x^2+\frac{l^2}{24}\right]$	=	$A_s l_x^2$	=	
A_{y^2}	$A_s\left[l_y^2+\frac{l^2}{24}\right]$	=	$A_s l_y^2$	=	$A_s\left[l_y^2+\frac{l^2}{24}\right]$	=	
A_{xy}	$A_s\left[l_x l_y+\frac{l^2}{24}\right]$	=	$A_s l_x l_y$	=	$A_s l_x l_y$	=	

GROUP Nº ___ FORM Nº 304 SHEET Nº ___

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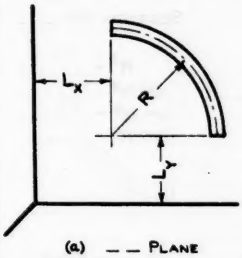
SUBJECT: COEFFICIENTS OF PIPE

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___



R = _____	R^2 = _____
.637 R = _____	.087 R^2 = _____
.664 R = _____	.091 R^2 = _____
1.806 R = _____	.095 R^2 = _____
K = _____	
1.571 KR = _____	

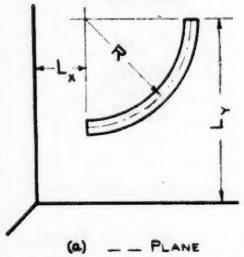
	(a) — — PLANE	(b) — — PLANE	(c) — — PLANE
L_X	_____	_____	_____
L_Y	_____	_____	_____
l_X	L_X + .637 R = _____	L_X + .664 R = _____	L_X = _____
l_Y	L_Y + .637 R = _____	L_Y = _____	L_Y + .664 R = _____
l_X^2	= _____	= _____	= _____
l_Y^2	= _____	= _____	= _____
$l_X l_Y$	= _____	= _____	= _____
A_S	1.571 KR = _____	1.806 R = _____	1.806 R = _____
A_X	$A_S l_X$ = _____	$A_S l_X$ = _____	$A_S l_X$ = _____
A_Y	$A_S l_Y$ = _____	$A_S l_Y$ = _____	$A_S l_Y$ = _____
A_{X^2}	$A_S[l_X^2 + .095R^2]$ = _____	$A_S[l_X^2 + .091R^2]$ = _____	$A_S l_X^2$ = _____
A_{Y^2}	$A_S[l_Y^2 + .095R^2]$ = _____	$A_S l_Y^2$ = _____	$A_S[l_Y^2 + .091R^2]$ = _____
A_{XY}	$A_S[l_X l_Y - .087R^2]$ = _____	$A_S l_X l_Y$ = _____	$A_S l_X l_Y$ = _____

GROUP Nº ___ FORM Nº 305 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS OF ___ PIPE

COMPUTED BY ____ DATE ____
CHECKED BY ____

SECTION Nº ___



R = ____	R^2 = ____	
.637 R = ____	.087 R^2 = ____	
.664 R = ____	.091 R^2 = ____	
1·806 R = ____	.095 R^2 = ____	
K = ____		
1·571 KR = ____		

	(a) ___ PLANE	(b) ___ PLANE	(c) ___ PLANE
L_x	____	____	____
L_y	____	____	____
l_x	L_x + .637 R = ____	L_x − .664 R = ____	L_x = ____
l_y	L_y − .637 R = ____	L_y = ____	L_y + .664 R = ____
l_x^2	= ____	= ____	= ____
l_y^2	= ____	= ____	= ____
$l_x l_y$	= ____	= ____	= ____
A_s	1·571 KR = ____	1·806 R = ____	1·806 R = ____
A_x	$A_s l_x$ = ____	$A_s l_x$ = ____	$A_s l_x$ = ____
A_y	$A_s l_y$ = ____	$A_s l_y$ = ____	$A_s l_y$ = ____
A_{x^2}	$A_s[l_x^2 + .095 R^2]$ = ____	$A_s[l_x^2 + .091 R^2]$ = ____	$A_s l_x^2$ = ____
A_{y^2}	$A_s[l_y^2 + .095 R^2]$ = ____	$A_s l_y^2$ = ____	$A_s[l_y^2 + .091 R^2]$ = ____
A_{xy}	$A_s[l_x l_y + .087 R^2]$ = ____	$A_s l_x l_y$ = ____	$A_s l_x l_y$ = ____

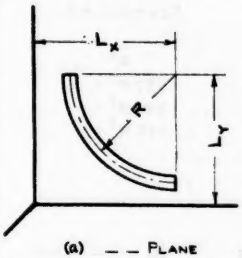
GROUP Nº ___ FORM Nº 306 SHEET Nº ____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS OF PIPE

COMPUTED BY
CHECKED BY

DATE

SECTION Nº



R = R^2 =
.637 R = .087 R^2 =
.664 R = .091 R^2 =
1.806 R = .095 R^2 =
K =
1.571 KR =

	(a) — — PLANE	(b) — — PLANE	(c) — — PLANE
L_x			
L_y			
l_x	$L_x - .637R$ =	$L_x - .664R$ =	L_x =
l_y	$L_y - .637R$ =	L_y =	$L_y - .664R$ =
l_x^2	=	=	=
l_y^2	=	=	=
$l_x l_y$	=	=	=
A_s	$1.571KR$ =	$1.806R$ =	$1.806R$ =
A_x	$A_s l_x$ =	$A_s l_x$ =	$A_s l_x$ =
A_y	$A_s l_y$ =	$A_s l_y$ =	$A_s l_y$ =
A_{x^2}	$A_s[l_x^2 + .095R^2]$ =	$A_s[l_x^2 + .091R^2]$ =	$A_s l_x^2$ =
A_{y^2}	$A_s[l_y^2 + .095R^2]$ =	$A_s l_y^2$ =	$A_s[l_y^2 + .091R^2]$ =
A_{xy}	$A_s[l_x l_y - .087R^2]$ =	$A_s l_x l_y$ =	$A_s l_x l_y$ =

GROUP Nº FORM Nº 307 SHEET Nº

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SUBJECT: COEFFICIENTS OF PIPE

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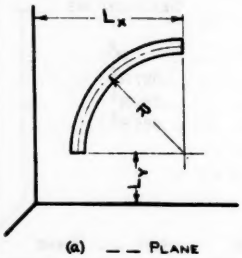

R = R^2 =
.637 R = .087 R^2 =
.664 R = .091 R^2 =
1.806 R = .095 R^2 =
K =
1.571 KR =

	(a) — — PLANE	(b) — — PLANE	(c) — — PLANE
L_x			
L_y			
l_x	L_x − .637 R =	L_x + .664 R =	L_x =
l_y	L_y + .637 R =	L_y =	L_y − .664 R =
l_x^2	=	=	=
l_y^2	=	=	=
$l_x l_y$	=	=	=
A_s	1.571 KR =	1.806 R =	1.806 R =
A_x	$A_s l_x$ =	$A_s l_x$ =	$A_s l_x$ =
A_y	$A_s l_y$ =	$A_s l_y$ =	$A_s l_y$ =
A_{x^2}	$A_s[l_x^2 + .095R^2]$ =	$A_s[l_x^2 + .091R^2]$ =	$A_s l_x^2$ =
A_{y^2}	$A_s[l_y^2 + .095R^2]$ =	$A_s l_y^2$ =	$A_s[l_y^2 + .091R^2]$ =
A_{xy}	$A_s[l_x l_y + .087R^2]$ =	$A_s l_x l_y$ =	$A_s l_x l_y$ =

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SUBJECT Coefficients Of Pipe

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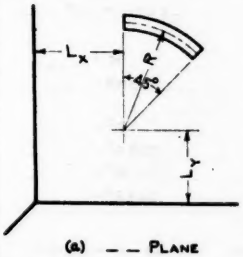

R = _____ K = _____
·373 R = _____ .785KR = _____
·382 R = _____
·828 R = _____ R^2 = _____
·900 R = _____ .008 R^2 = _____
·904 R = _____ .017 R^2 = _____
·978 R = _____ .043 R^2 = _____

	(a) — — Plane	(b) — — Plane	(c) — — Plane
L_x	_____	_____	_____
L_y	_____	_____	_____
l_x	L_x + ·373 R = _____	L_x + ·904 R = _____	L_x = _____
l_y	L_y + ·900 R = _____	L_y = _____	L_y + ·382 R = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	·785 KR = _____	·978 R = _____	·828 R = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A_s[l_x^2 + .043R^2]$ = _____	$A_s[l_x^2 + .008R^2]$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A_s[l_y^2 + .008R^2]$ = _____	$A_s l_y^2$ = _____	$A_s[l_y^2 + .043R^2]$ = _____
A_{xy}	$A_s[l_x l_y - .017R^2]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

GROUP Nº ____ FORM Nº 309 SHEET Nº _____
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SUBJECT Coefficients of Pipe

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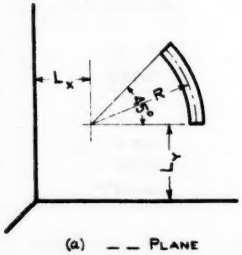

R = _____ K = _____
.373 R = _____ .785KR = _____
.382 R = _____
.828 R = _____ R^2 = _____
.900 R = _____ .008 R^2 = _____
.904 R = _____ .017 R^2 = _____
.978 R = _____ .043 R^2 = _____

	(a) — — Plane	(b) — — Plane	(c) — — Plane
L_x	_____	_____	_____
L_y	_____	_____	_____
l_x	L_x + .900R = _____	L_x + .382R = _____	L_x = _____
l_y	L_y + .373R = _____	L_y = _____	L_y + .904R = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	.785KR = _____	.828R = _____	.978R = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A_s[l_x^2 + .008R^2]$ = _____	$A_s[l_x^2 + .043R^2]$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A_s[l_y^2 + .043R^2]$ = _____	$A_s l_y^2$ = _____	$A_s[l_y^2 + .008R^2]$ = _____
A_{xy}	$A_s[l_x l_y - .017R^2]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

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SUBJECT Coefficients Of Pipe

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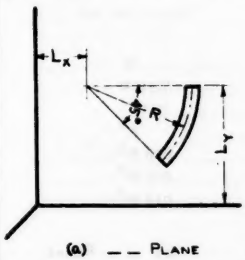

R = _____ K = _____
.373 R = _____ .785KR = _____
.382 R = _____
.828R = _____ R^2 = _____
.900R = _____ $.008R^2$ = _____
.904R = _____ $.017R^2$ = _____
.978R = _____ $.043R^2$ = _____

	(a) ___ Plane	(b) ___ Plane	(c) ___ Plane
L_x	_____	_____	_____
L_y	_____	_____	_____
l_x	$L_x + .900R$ = _____	$L_x - .382R$ = _____	L_x = _____
l_y	$L_y - .373R$ = _____	L_y = _____	$L_y + .904R$ = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	.785KR = _____	.828R = _____	.978R = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A_s[l_x^2 + .008R^2]$ = _____	$A_s[l_x^2 + .043R^2]$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A_s[l_y^2 + .043R^2]$ = _____	$A_s l_y^2$ = _____	$A_s[l_y^2 + .008R^2]$ = _____
A_{xy}	$A_s[l_x l_y + .017R^2]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

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SUBJECT COEFFICIENTS OF PIPE

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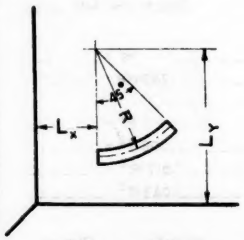

R = _____	K = _____
.373 R = _____	.785KR = _____
.392 R = _____	
.828 R = _____	R^2 = _____
.900 R = _____	.008 R^2 = _____
.904 R = _____	.017 R^2 = _____
.978 R = _____	.043 R^2 = _____

	(a) __ PLANE	(b) __ PLANE	(c) __ PLANE
L_x	_____	_____	_____
L_y	_____	_____	_____
l_x	L_x+.373R = _____	L_x−.904R = _____	L_x = _____
l_y	L_y−.900R = _____	L_y = _____	L_y+.392R = _____
l_x^2	= _____	= _____	= _____
l_y^2	= _____	= _____	= _____
$l_x l_y$	= _____	= _____	= _____
A_s	.785KR = _____	.978R = _____	.828R = _____
A_x	$A_s l_x$ = _____	$A_s l_x$ = _____	$A_s l_x$ = _____
A_y	$A_s l_y$ = _____	$A_s l_y$ = _____	$A_s l_y$ = _____
A_{x^2}	$A_s[l_x^2+.043R^2]$ = _____	$A_s[l_x^2+.008R^2]$ = _____	$A_s l_x^2$ = _____
A_{y^2}	$A_s[l_y^2+.008R^2]$ = _____	$A_s l_y^2$ = _____	$A_s[l_y^2+.043R^2]$ = _____
A_{xy}	$A_s[l_x l_y+.017R^2]$ = _____	$A_s l_x l_y$ = _____	$A_s l_x l_y$ = _____

GROUP Nº ____ FORM Nº 312 SHEET Nº _____
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SUBJECT Coefficients of Pipe

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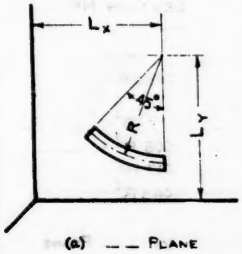
SECTION No. ___



R =		K =	
.373R =		.785KR =	
.382R =			
.828R =		R^2 =	
.900R =		.008 R^2 =	
.904R =		.017 R^2 =	
.978R =		.043 R^2 =	

	(a) — — Plane	(b) — — Plane	(c) — — Plane
L_x			
L_y			
l_x	$L_x - .373R$ =	$L_x - .904R$ =	L_x =
l_y	$L_y - .900R$ =	L_y =	$L_y - .382R$ =
l_x^2	=	=	=
l_y^2	=	=	=
$l_x l_y$	=	=	=
A_s	.785KR =	.978R =	.828R =
A_x	$A_s l_x$ =	$A_s l_x$ =	$A_s l_x$ =
A_y	$A_s l_y$ =	$A_s l_y$ =	$A_s l_y$ =
A_{x^2}	$A_s[l_x^2 + .043R^2]$ =	$A_s[l_x^2 + .008R^2]$ =	$A_s l_x^2$ =
A_{y^2}	$A_s[l_y^2 + .008R^2]$ =	$A_s l_y^2$ =	$A_s[l_y^2 + .043R^2]$ =
A_{xy}	$A_s[l_x l_y - .017R^2]$ =	$A_s l_x l_y$ =	$A_s l_x l_y$ =

GROUP No. ___ FORM No 313 SHEET No. ___

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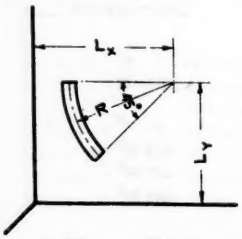

R = ___ K = ___

·373 R = ___ .785KR = ___

·382 R = ___

·828 R = ___ R^2 = ___

·900 R = ___ .008 R^2 = ___

·904 R = ___ .017 R^2 = ___

·978 R = ___ .043 R^2 = ___

	(a) ___ PLANE	(b) ___ PLANE	(c) ___ PLANE
L_x	___	___	___
L_Y	___	___	___
l_x	L_x − ·900R = ___	L_x − ·382 R = ___	L_x = ___
l_Y	L_Y − ·373 R = ___	L_Y = ___	L_Y − ·904 R = ___
l_x^2	= ___	= ___	= ___
l_Y^2	= ___	= ___	= ___
$l_x l_Y$	= ___	= ___	= ___
A_s	.785 KR = ___	.828 R = ___	.978 R = ___
A_x	$A_s l_x$ = ___	$A_s l_x$ = ___	$A_s l_x$ = ___
A_Y	$A_s l_Y$ = ___	$A_s l_Y$ = ___	$A_s l_Y$ = ___
A_{x^2}	$A_s[l_x^2 + .008R^2]$ = ___	$A_s[l_x^2 + .043R^2]$ = ___	$A_s l_x^2$ = ___
A_{Y^2}	$A_s[l_Y^2 + .043R^2]$ = ___	$A_s l_Y^2$ = ___	$A_s[l_Y^2 + .008R^2]$ = ___
A_{xY}	$A_s[l_x l_Y - .017R^2]$ = ___	$A_s l_x l_Y$ = ___	$A_s l_x l_Y$ = ___

GROUP Nº ___ FORM Nº 314 SHEET Nº ___

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SUBJECT: COEFFICIENTS OF ___ PIPE ___

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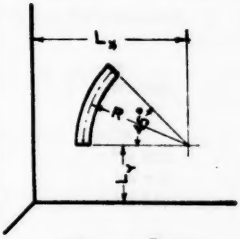
SECTION Nº ___



R = ___ K = ___
·373 R = ___ .785KR = ___
·382 R = ___
·828 R = ___ R^2 = ___
·900 R = ___ .008 R^2 = ___
·904 R = ___ .017 R^2 = ___
·978 R = ___ .043 R^2 = ___

	(a) ___ PLANE	(b) ___ PLANE	(c) ___ PLANE
L_x	___	___	___
L_y	___	___	___
l_x	L_x − .900R = ___	L_x + .382 R = ___	L_x = ___
l_y	L_y + .373 R = ___	L_y = ___	L_y − .904 R = ___
l_x^2	= ___	= ___	= ___
l_y^2	= ___	= ___	= ___
$l_x l_y$	= ___	= ___	= ___
A_s	.785KR = ___	.828 R = ___	.978 R = ___
A_x	$A_s l_x$ = ___	$A_s l_x$ = ___	$A_s l_x$ = ___
A_y	$A_s l_y$ = ___	$A_s l_y$ = ___	$A_s l_y$ = ___
A_{x^2}	$A_s[l_x^2 + .008R^2]$ = ___	$A_s[l_x^2 + .043R^2]$ = ___	$A_s l_x^2$ = ___
A_{y^2}	$A_s[l_y^2 + .043R^2]$ = ___	$A_s l_y^2$ = ___	$A_s[l_y^2 + .008R^2]$ = ___
A_{xy}	$A_s[l_x l_y + .017R^2]$ = ___	$A_s l_x l_y$ = ___	$A_s l_x l_y$ = ___

GROUP Nº ___ FORM Nº 315 SHEET Nº ___
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS OF ___ PIPE ___

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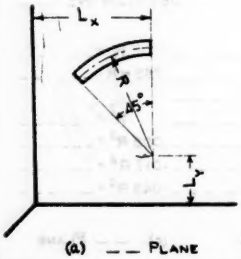
SECTION Nº ___



R = ___ K = ___

.373 R = ___ 785KR = ___

.382 R = ___

.828 R = ___ R^2 = ___

.900 R = ___ .008 R^2 = ___

.904 R = ___ .017 R^2 = ___

.978 R = ___ .043 R^2 = ___

	(a) ___ PLANE	(b) ___ PLANE	(c) ___ PLANE
L_x	___	___	___
L_y	___	___	___
l_x	$L_x - .373R$ = ___	$L_x + .904R$ = ___	L_x = ___
l_y	$L_y + .900R$ = ___	L_y = ___	$L_y - .382R$ = ___
l_x^2	= ___	= ___	= ___
l_y^2	= ___	= ___	= ___
$l_x l_y$	= ___	= ___	= ___
A_s	$.785KR$ = ___	$.978R$ = ___	$.828R$ = ___
A_x	$A_s l_x$ = ___	$A_s l_x$ = ___	$A_s l_x$ = ___
A_y	$A_s l_y$ = ___	$A_s l_y$ = ___	$A_s l_y$ = ___
A_{x^2}	$A_s[l_x^2 + .043R^2]$ = ___	$A_s[l_x^2 + .008R^2]$ = ___	$A_s l_x^2$ = ___
A_{y^2}	$A_s[l_y^2 + .008R^2]$ = ___	$A_s l_y^2$ = ___	$A_s[l_y^2 + .043R^2]$ = ___
A_{xy}	$A_s[l_x l_y + .017R^2]$ = ___	$A_s l_x l_y$ = ___	$A_s l_x l_y$ = ___

GROUP Nº ___ FORM Nº 316 SHEET Nº ___

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NOTES.

INDEX TO NOTES.

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MECHANICAL PROPERTIES OF SOLIDS.

The 1941 James Forrest Lecture of the Institution of Civil Engineers was delivered by Professor E. N. da.C. Andrade. Professor Andrade has been Quain Professor of Physics in the University of London since 1928, Scientific Adviser to the Director of Scientific Research, Ministry of Supply, since the outbreak of the European War. For several years he was Professor of Physics in the Royal Artillery College at Woolwich. He was also Editor for Physics of the Encyclopedia Brittanica. Professor Andrade states that the strength of solids is not nearly so high as it should be; in fact, under test it is only about one-thousandth of that determined theoretically. He further suggests the present iron and steel age will soon be passed, just as civilization has already passed through the stone and brass ages. His lecture is reprinted from the June, 1941, Journal of the Institution of Civil Engineers.

INTRODUCTION.

It is a strange paradox of science that the more gross and palpable the phenomena, the harder it is to win an understanding as to their true nature; that the ultimate mechanism of happenings which involve objects that we can see and handle readily is much more obscure and complicated than that of things concerning which our knowledge is only indirect. Scientifically the behavior of gases is much more readily explained than that of liquids or solids, although man as an artificer has been dealing with the latter for hundreds, nay thousands, of years, and had amassed isolated rules as to individual behavior before he knew of the existence of any gas but air. With gases themselves we find that the more tenuous they are, and in that sense the more difficult to handle, the simpler they are. The flow of gases at low pressure is simple, that of liquids more difficult, that of solids still more complicated. The flow of electricity through metals, the laws of which were worked out long before the flow of electricity through gases was studied seriously, offers much greater difficulties to the theoretical physicist than does the flow of electricity through rarefied gases.

The reason for this contrast is, of course, simple. It is easier to explain the behavior of atoms and molecules, in terms of which our modern theories are constructed, when they are widely separated, so that for most of the time their action upon one another is negligible, than when they are packed close together, so that everything depends upon the forces which they exert upon one another.

The physicist who is interested in the theoretical and the predictable behavior of electricity and the modern electrical engineer meet upon the common ground of electronic studies, which have today such important technical applications. The civil engineer and the physicist who is interested in the fundamental behavior of matter, the atomic explanation of its mechanical properties, have so far scarcely come into touch, because precisely those properties of solids which confront the builder of bridges and the designer of dams—and which have, in some cases, confronted him for centuries—are those which the theorist finds the most intractable. Any craftsman can show the most experienced physicist things about the behavior of iron, for instance, which it will puzzle him to explain. A piece of copper wire bent around the finger forms a smooth curve, whilst a piece of soft iron wire of the same dimensions makes sharp bends and takes polygonal form. Why? Orowan[1], who called attention to the phenomenon, has put forward a tentative explanation, it is true, but it only throws the matter back on phenomena that are still only partially understood.

[1] Proc. Phys. Soc., 52, 8, 1940.

The Mechanical Behavior of Solids.

What I want to discuss today is the mechanical behavior of solids. I have already pointed out that the difficulty of furnishing a molecular explanation for this behavior lies in the fact that, with solids, the proximity of the molecules makes the intermolecular forces of prime importance. In liquids the molecules are packed together about as densely as they are in solids, but there is one great and familiar difference between solids and liquids that is fundamental for our subject. With a liquid of given chemical composition the physical properties are perfectly definite and can be measured on different specimens with a consistency limited only by the precision of the measurements; with a solid of given chemical nature, say a pure metal, many of the properties may vary widely from specimen to specimen, according to the previous history of that specimen. In particular, the mechanical strength is not accurately specified when we have specified the substance and its degree of purity, and, as a further case of solid idiosyncrasy, certain rather complicated properties of ionic crystals depend markedly on the state of the specimen. Even the density depends somewhat on the treatment; thus while, in the Tables, the density of mercury is given to within 1 part in 100,000, the values for the density of copper vary over a range of 4 per cent.

Some properties, however, of solids are quite consistent from specimen to specimen: we may cite specific heat, melting-point, coefficient of thermal expansion and lattice constants. Smekal[1] has called the first class of properties, those, like mechanical strength in particular, which vary widely from specimen to specimen, according to the history of the sample, "structure-sensitive," as against the "structure-insensitive" properties just cited. The existence of structure-sensitive properties in crystals shows one thing clearly, that actual crystals cannot consist of atoms or molecules arranged in the perfect pattern contemplated by the mathematician—on a perfect lattice, to speak in more technical language—nor can they be structures in thermodynamic equilibrium. If the crystals were perfect they would all have the same properties if of the same material; similarly, if they were in thermodynamic equilibrium, they would eventually, from whatever arrangement they started, reach a final arrangement corresponding to the least free energy, and this arrangement, and consequently the mechanical properties, would always be the same. All crystals as usually dealt with must, then, be imperfect, although the possibility of preparing a perfect crystal must not be definitely excluded. The essential difference for our purposes is that a solid can support internal stresses, which may be different from specimen to specimen, while a liquid, by definition, cannot support any internal stresses permanently.

The Behavior of Crystals.

Let us consider how a perfect crystal should behave under stress. Each atom is in equilibrium because both attractive and repulsive forces exist between any two atoms, varying inversely as different powers of the distance between them—say an attractive force varying as $1/r^2$ and a repulsive force varying as $1/r^n$, where n may be somewhere about 10. At the equilibrium distance these forces just counterbalance; if we try to separate the atoms the attractive force preponderates, while if we try to push them together the repulsive force preponderates. If we try to slide one plane of atoms over another plane, or, say, one row of atoms over another, the distance between two adjacent atoms first increases and then decreases again; the force will rise to a maximum and then decrease. If, therefore, we apply an increasing tension normal to a given set of crystal planes, the distances between the atoms should increase until the tension is sufficient to overcome

[1] Phys. Z., 34, 633, 1933.

the attractive forces, when the planes should separate and the crystal should break. Until this point is reached the crystal should, if we release the tension, always return to its original state. We should expect a range of perfect elasticity, followed by sudden fracture. For a shearing stress we should likewise have a region of perfect elasticity, but if a stress be sufficient to cause the plane to move one-half of one atomic step with reference to the neighboring plane, then it should be sufficient to cause it to execute any number of steps. A shearing stress exceeding a certain value would produce indefinite glide of one plane over the other, without, of course, any hardening.

Actually, as we shall see, nothing of this kind occurs. Before passing on, however, we will inquire how a rough theoretical estimate of mechanical strength can be made. When we break an ideal crystal rod in tension two fresh surfaces are formed, and if the broken pieces of crystal are identical in structure with the original crystal the only increase in energy will be represented by the energy of these surfaces. This must be derived from, and cannot exceed, the elastic energy stored in the crystal at the moment of breaking: if F is the breaking stress, d the extension at breaking-point of a length equal to the intermolecular distance a, γ the surface tension, and if we consider unit cross-sectional area, then, if Hooke's law holds up to fracture,

$$\text{work done} = \tfrac{1}{2}Fd \geq 2\gamma;$$

and if E is Young's modulus,

$$F = \frac{d}{a}E,$$

so that

$$F^2 \geq \frac{4\gamma E}{a}$$

To find F we have to estimate the surface tension of a solid. This cannot be measured by ordinary methods, but, since the intermolecular distances concerned in the solid state are not very different from those in the liquid state, the intermolecular forces must be much the same in both cases and the surface tension of a substance in the solid state cannot be very different from that of the same substance in the liquid state. For liquid metals the surface tensions are of the order of 500 dynes per centimeter[1], the intermolecular distance is about 3×10^{-8} centimeter and Young's modulus, for the stronger metals about 2 megabars[2]. This gives for F a value of about 360 kilobars (= 2300 tons weight per square inch), which is about forty times the tensile strength of the toughest metal[3]. If we take the yield-point,

[1] The recorded figures vary from 350 dynes per centimeter for antimony at 640 degrees C. to 1150 dynes per centimeter for a certain cast iron at 1150 degrees C. for mercury at 20 degrees C. the figure is 465 dynes per centimeter.

[2] Strength is expressed in various units, such as tons weight per square inch; pounds weight per square inch; grams weight per square centimeter; grams weight per square millimeter; kilograms weight per square millimeter ("weight" being habitually, but unfortunately omitted); dynes per square millimeter, and so on. The best unit for scientific work would seem to be the *bar*, which is 10^6 dynes per square centimeter: it is already habitually used by the meteorologists to express pressure. 10 grams weight per square millimeter is (within 2 per cent) 1 bar; 1 ton weight per square inch is 154-6 bars. The kilobar is, of course, a thousand bars: the megabar a million bars. I hope that engineers will consider the gradual adoption of the system. 1 kilobar is about 6-5 tons weight per square inch; 1 bar is about 14.5 pounds weight per square inch.

[3] I prefer not to use the term "ultimate tensile strength," since it is usually employed in a very unfortunate way by engineers. They ordinarily denote by it the breaking load divided by the original cross-sectional area, whereas, except in the unusual case of brittle fracture, marked reduction of area ("necking") precedes rupture. The term, used in this conventional manner, has no significance, and its use has led to much confusion. In the case of plastic flow of metals I showed some time ago the great simplification that results from measuring the flow under constant stress instead of under constant load.

instead of the tensile strength for comparison with the theoretical value, the discrepancy is even greater. Similar calculations can be carried out for the theoretical shear strength, and give a value of about 130 kilobars, some forty times that observed for steel. Thus even the toughest metals show much less than the theoretical strength.

The ordinary metals, however, are polycrystalline metals; it may be suggested that for comparison with theory it would be fairer, perhaps, to consider a single crystal. I take rock salt as an example, for it is a simple face-centered cubic crystal composed of alternate positive and negative ions, for which the mathematicians can carry out exact calculations, of, for example, the theoretical strength. For rock salt this works out to be 20 kilobars and rupture should theoretically be preceded by an elastic extension of 14 per cent. It is interesting to note that surface-tension considerations give a similar value of the theoretical strength, which confirms our belief in the validity of the surface-tension method of estimating strength. The actual breaking strength of rock salt under tension is, however, about 20 bars, that is, about one-thousandth of what it should be. Rock salt does give brittle fracture, with very small extension, at ordinary temperatures, although at higher temperatures, above 600 degrees C., it shows plastic behavior.

A perfect ionic crystal of a type which is well understood theoretically is, then, in practice much too weak and has much too small an elastic region. Let us look at its behavior in a little more detail, for it will help us later with our metals. To examine a transparent crystal we have an agent which we cannot use for metals, polarized light. A cubic crystal such as rock salt has, in an unstrained condition, no effect on such light; any strain that makes the crystal depart from its precise cubic structure causes it, however, to become doubly refracting. Obreimow and Schubnikow[1] loaded a crystal of rock salt in a beam of polarized light; at a low stress, of about 8 bars, bright streaks appeared along two directions at right angles, which were the traces of (110) planes. These lines did not appear as soon as the stress was applied, but arose spontaneously some 20 seconds later. As the stress was increased the bright lines increased in number, but appeared always in the same crystallographic directions. Surface marks also appear in the same direction if the crystal undergoes plastic extension at high temperature. The crystal shows permanent slip on certain crystallographic planes, but the polarized light shows that its crystalline perfection has been destroyed along these planes. The glide-planes were established much earlier in metals, but I quote rock salt first because of the ease with which this latter point can be established. The same thing has been shown with other transparent crystals, for example, potassium halides.

Glass: Surface Effects.

Let us see if we fare any better as regards strength if we take a non-crystalline substance, say glass. The surface tensions of molten glasses are round about 150 dynes per centimeter, which gives theoretical strengths of the order 100 kilobars. The actual strength of ordinary glass fibers, which give brittle fracture at room temperatures, is from 300 to 900 bars —less than one-hundredth of what it should be. Here, however, we meet a curious fact of great importance for our subject: freshly drawn glass fibers, of hard glass, are very much stronger than old fibers, of the same glass, but gradually lose their strength as time goes on and come down to a steady value. Touching or handling the fresh fibers weakens them. A. A. Griffith[2], who has carried out fundamental work on this subject, assumes

[1] Z. Physik, 41, 907, 1927.
[2] Phil. Trans. Roy. Soc., A 221, 163, 1921.

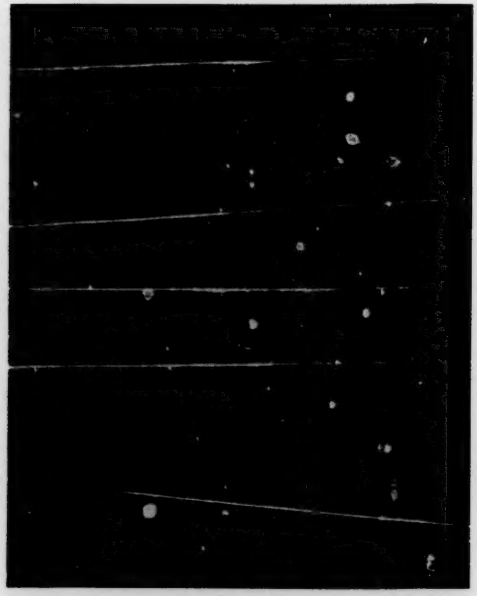
that the cause of the weakening is the formation of invisible surface cracks, which are absent in a fresh fire-drawn material. The breaking strength will depend upon the depth of the crack and the radius of curvature at the end: the greater the former and the smaller the latter, the higher the local stress and the weaker the material. There is a limit to the radius of curvature: it cannot be less than something of the order of the interatomic distance. If we assume a reasonable value we can work out the depth of the crack needed to give the observed strength, and it comes out about 1 or 2μ, say twice the wavelength of visible light. Of course if we have threads of this order of diameter we cannot have cracks of this depth, and very fine threads are, in fact, stronger per unit cross-section than large threads. The variations of stress, due to cooling, necessary for the formation of the cracks probably cannot be established in such fine threads.

These cracks are no longer a mere hypothesis. Mr. Tsien and I[1] found that attack by sodium vapor (but not by hydrofluoric acid) would develop the cracks and make them visible. On freshly drawn hard glass tube no, or very few, cracks could be detected—on the same tube, when old, many cracks were made visible. They ran transverse to the direction of drawing of the glass rod or tube, as they should. Various controls established that we were not dealing with a spurious phenomenon. Figure 1 shows typical cracks, developed by sodium vapor, on the inside of a pyrex glass tube: Figure 2 shows the same kind of thing with quartz glass. Dr. Martindale and I[2] also found that when very thin films of gold and silver were heated to about 300 degrees C. on glass surfaces minute crystallites formed up in lines, which were in general parallel. These lines are not caused by visible scratches but are attributable to minute surface cracks, a hypothesis which is confirmed by the behavior of the two known types of diamond, one of which is perfect while the other contains invisible flaws, made evident by certain physical properties. The perfect type shows no lines of crystallites; with the other type such lines form. I may add that just before the war I succeeded in showing the minute surface cracks in hard glass by a special type of illumination[3].

An experiment of Orowan's[4] is instructive in this connection. If a strip of mica be loaded by means of parallel metal clamps, extended across the whole strip, so that the edges are stressed, we find a maximum strength of about 3 kilobars, approximately that of wrought iron and very much in excess of that of glass and of rock salt. The flat faces of mica are very perfect[5], but the cut edges have, of course, no such perfection. By using special clamps, gripping the sheet near the middle only, stress at these edges can be avoided. For a mica sheet loaded in this way the strength goes up to 32 kilobars, more than ten times that with the other method of loading, and much in excess of that of the best constructional steel. The prejudicial effect of trivial and invisible cracks at the cut edge is clear.

These considerations furnish us with fundamental material for the understanding of brittle rupture, but possibly appear very remote from any practical consideration. They may, however, not always be so. Recently a serious proposal to use ordinary glass as a substitute for steel in reinforced concrete has been put forward by Messrs. Soden and Lincoln[6]. The glass is introduced in the form of flat "planks" set on edge in the beam, as shown in Figure 3, the lower edges being in tension and the upper edges unstressed or in slight compression[7]. Now the inventors specify that the lower edge

[1] Proc. Roy. Soc., A 159, 346, 1937.
[2] Phil. Trans. Roy. Soc., A. 235, 69, 1935.
[3] Unpublished.
[4] Z. Physik, 82, 235, 1933.
[5] They show no lines of crystallites when metal films are heated on them.
[6] Engineering, 150, 225, 1940.
[7] The inventors take for the safe stress in glass 170 bars (2500 pounds per square centimeter), about one-half of the worst figure quoted for tensile strength and one-fifth of the best—the tensile strength of the particular glass used is not given.



× 80

FIGURE 1.—TYPICAL SURFACE CRACKS IN PYREX GLASS, DEVELOPED BY SODIUM VAPOR.

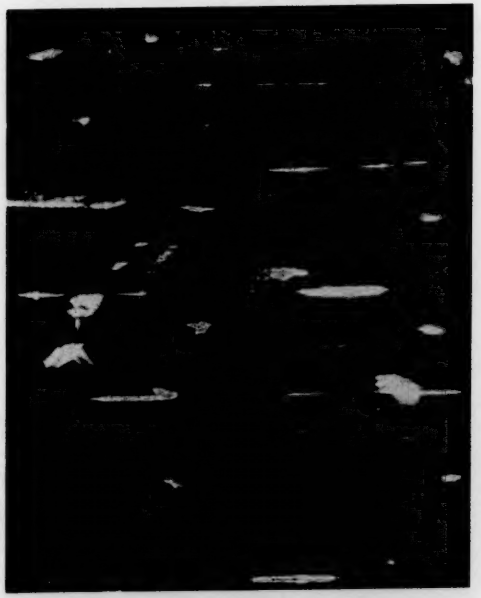


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FIGURE 2.—TYPICAL CRACKS IN QUARTZ GLASS, DEVELOPED BY SODIUM VAPOR.

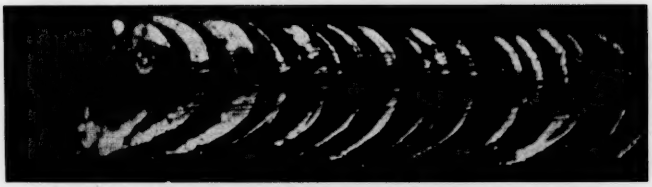

FIGURE 5.—SLIP PLANES OF MOLYBDENUM AT 1000 DEGREES C.

FIGURE 6.—SLIP PLANES OF MOLYBDENUM AT 1500 DEGREES C.

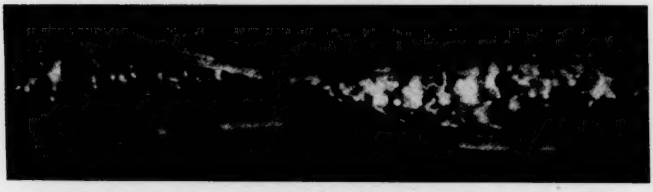

FIGURE 7.—SLIP PLANE OF MOLYBDENUM AT 2000 DEGREES C.

FIGURE 8.—SLIP PLANES OF CADMIUM AT ROOM TEMPERATURE

must be a fire-finished edge, while the upper edge, which is cut, must not be in tension. The minute cracks in a cut edge are certainly much more pronounced than those in a fire-finished edge. Glass-reinforced beams have shown themselves perfectly satisfactory under static load, but offer some disadvantages under impact test, as compared with ordinary reinforced concretes. You will not expect me to pronounce definitely as to the feasibility of utilizing this new invention, but the proposal is evidently worthy of serious consideration.

There are other industrial examples of the importance of the avoidance of surface cracks. Toughened glass is prepared by rapidly cooling the surface of the slabs while the interior is still molten. When the internal part solidifies and cools the shrinkage pulls the surface into a state of compression, which prevents the formation of cracks. The glazing of industrial porcelain, for insulators and such like, is also instructive in this connection.

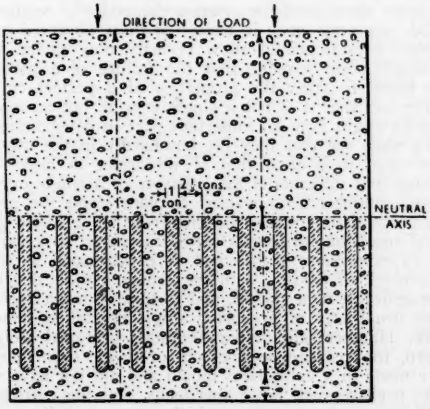

FIGURE 3.—CONCRETE REINFORCED BY GLASS.

If the object is to be strong the glaze must have a coefficient of thermal expansion lower than that of the porcelain body. When the cooling takes place after glazing, the greater shrinkage of the body pulls the surface glaze into a state of compression. If a glaze with a higher coefficient of expansion be used, the strength of the porcelain is less than that of the unglazed body. The swelling of the metal surface by the process of nitriding may also act as a talisman against fatigue cracks, which usually start at the surface.

The study of ionic crystals, then, has shown us that yield takes place along certain crystallographic planes, and that along these planes a state of permanent internal stress is set up. The study of glasses and of mica has shown us that submicroscopic inperfections, invisible cracks, play a fundamental part in the yield with brittle fracture. I may add that a certain peculiar behavior of rock salt has been invoked to support the theory that surface cracks are the origin of glide. If a plate of rock salt be held under warm water it can be bent like a piece of lead, and in general shows a plastic behavior which contrasts strongly with its brittle behavior under

ordinary conditions. Joffe[1], who directed attention to the effect, attributes it to the solution of the surface layers and, in consequence, the removal of surface cracks; others have explained it as due to the penetration of water into the crystal structure. I have shown[2], however, that a crystal can be readily bent under cold running water; this, owing to the avoidance of any approach to saturation, will give a high rate of surface solution, but a smaller diffusion than warm water, and so speaks for a surface effect.

The Mechanical Behavior of Metals.

I now turn to the question of the mechanical behavior of metals. Metals, as the engineer knows them, consist of masses of crystals, separated by boundaries that appear sharp under the highest magnification, and the study of the crystal structure of metals, initiated 77 years ago by Sorby, is well known to be of the greatest importance for the metallurgist. As we have seen, the behavior of a perfect single crystal under stress should theoretically be either brittle fracture, within the elastic limit, under tension, or unlimited glide under shear, and the stresses theoretically required to produce fracture or glide are very much in excess of those observed experimentally. Further, an ideal crystal should show no hardening under plastic flow.

At first it might seem possible that the failure of a piece of ordinary polycrystalline metal to show the properties of a perfect crystal could be attributed to the fact that it is not a single crystal, but an aggregate. The crystal boundaries might play an important part in determining the metallic properties. We shall see that they do, but hardly in the way to be anticipated. In any case, it is clear that if we desire to understand metallic behavior we must first understand the behavior of a single crystal of metal, just as any civil engineer who wanted to understand the properties of a girder structure would first study the properties of a single girder.

Single crystal specimens of metals can be prepared in a variety of ways. If one desires to experiment with a rod of crystal such as rock salt, it is usual to take a block of crystal, with the ordinary faces developed, so that it is clearly recognizable as a crystal, and to turn or cut a specimen from it. This is not the usual way of preparing single crystal specimens of metal, although K. W. Hausser[3] did prepare crystals of metal weighing several pounds, and turn, for instance, spheres out of them; even then the original specimens were made in a cylindrical mould and did not have natural faces. It is difficult to prepare such large specimens, and the cutting of the test-piece is likely to introduce strains which, as we shall see, modify the properties. The better and more usual way is to prepare the specimen from ordinary polycrystalline metal, and then, by suitable treatment, to change its inner structure to that of one crystal, leaving the outer faces unmodified. Sir Harold Carpenter and Miss Elam[4], for instance, cut flat test-pieces of aluminum, of the ordinary form, and then by a process of repeated strain and reheating converted the whole piece into one crystal, or occasionally into two or three crystals with boundaries. The more usual form, however, is that of a cylindrical rod or stout wire.

I prepared the first single crystal specimens of certain metals by the suitable heating of wires as long ago as 1913; but the work was interrupted by the War of 1914 and it was not until nearly 20 years later that I was able to take up the subject again. A usual method is to cause the desired crystallization by melting the wire and then, by suitable progressive cooling from one end, promote the growth of the crystal structure from a single small "seed" crystal. There are many modifications of the method, with which I must not trouble you now.

[1] Z. Physik, *22*, 286, 1924.
[2] Unpublished.
[3] Wiss. Veroff. Siemens-Konzern, 5, 144, 1927.
[4] Proc. Roy. Soc. A, 100, 329, 1921.

The specimen so prepared will, to casual inspection, be indistinguishable from a wire or rod of ordinary metal. X-rays, of course, will at once reveal its nature, as all the reflecting planes, of one kind, will be parallel (or very nearly parallel) to one another. There is no need, however, to use X-rays, valuable as they are as a method of investigation, to show that a single crystal specimen is not an ordinary wire; the simplest mechanical tests will at once give evidence that there is something peculiar about it.

In the first place metal single crystals are not, as might be expected, very strong but they are very soft. They also show very remarkable strain hardening, a phenomenon which I have not hitherto discussed. A copper crystal rod half an inch in diameter behaves like lead at first manipulation; it can be very easily bent in the hands, but becomes progressively harder to deform. It takes a very strong man to bend back to its original form such a rod that has once been deformed into a semicircle. Brittle behavior is not unknown in single crystal specimens (bismuth crystals, for instance, show brittle fracture at low temperature), but it is not typical.

The Geometry of Glide.

Remarkable properties of single crystal wires can be exhibited by simply subjecting them to tension. On slight extension a number of parallel markings appear on the surface and as elongation proceeds these become, in general, more and more marked: in some cases the number increases. The wire retains practically its full diameter in one direction and thins in a direction more or less normal to this, so that in the case of wires that can be pulled to several times their original length, such as, for example, cadmium wires, the original cylinder becomes a thin ribbon. As the result of the work of Polanyi, Schmidt, G. I. Taylor, and others this behavior has been expressed in terms of the geometry of the crystal.[1] The slip takes place preferentially on certain types of crystal planes and in certain definite crystallographic directions—the glide planes and glide directions.

Conditions are simplest in the metals of hexagonal crystal structure, for here the usual glide plane[2] is the hexagonal base, which is unique. In face-centered cubic crystals, on the other hand, glide takes place on the octahedral faces, of which there are four, all exactly equivalent to one another. Considering, then, for the moment hexagonal metals, such as zinc, cadmium and magnesium, the slip takes place on one set of planes, all parallel to one another. As regards direction, it takes place along a line joining opposite corners of the hexagon, so that there are three equivalent glide directions. Which of these three is effective depends upon the direction in which the force is applied. Figure 4[3] illustrates this schematically: (*a*) shows the wire before slip, the plane exposed being structurally a hexagonal basal plane, with the directions of the three digonal axes indicated by the hexagon. The digonal axis nearest to the direction T of the tension, projected on the plane, will be the one along which glide takes place; (*c*) shows the situation after glide; (*b*) and (*d*) are views of (*a*) and (*c*) respectively, seen from a line in the glide plane and normal to the wire axis.

If slip took place equally readily on all parallel atomic planes the crystal surface would, of course, be as smooth after extension as it is before extension. What takes place, however, is a preferential glide in the neighborhood of certain planes, the so-called slip planes, spaced at more or less regular intervals, so that the crystal appears to slip in parallel slices or slabs. This produces the remarkable stepped appearance characteristic of crystal wires that have been much extended. In general the spacing of the slip planes becomes wider and wider as the temperature is raised. With

[1] See, for example, C. F. Elam, "Distortion of Metal Crystals," Oxford, 1935.

[2] Under certain conditions of temperature glide can take place on other planes, the so-called prismatic faces of the first and second kind.

[3] Adapted from E. Schmid and W. Boas, Kristallplastizitat, A 59, Berlin, 1935.

molybdenum at 1000 degrees C. and 1500 degrees C., for instance, the coarse appearance of the slip packets is very striking, as seen in Figures 5[1] and 6[2]; with molybdenum at 2000 degrees C. the slip may take place all at one plane as shown in Figure 7[2]. This may be compared with the appearance of cadmium at room temperature, Figure 8[3]. It has been suggested that the choice of the particular region where marked glide takes place may be due to local impurities, but experiments carried out in my laboratory with wires of exceedingly pure solid mercury[4] (the impurity was probably about 1 in a hundred million, show that this cannot be so. The slip bands are very fine and close. Figures 9[5] and 10[6] show photographs, due to Dr. Greenland, of slip in wires of solid mercury, the former viewed from a direction in the glide plane and normal to the wire axis. The whole question of the forma-

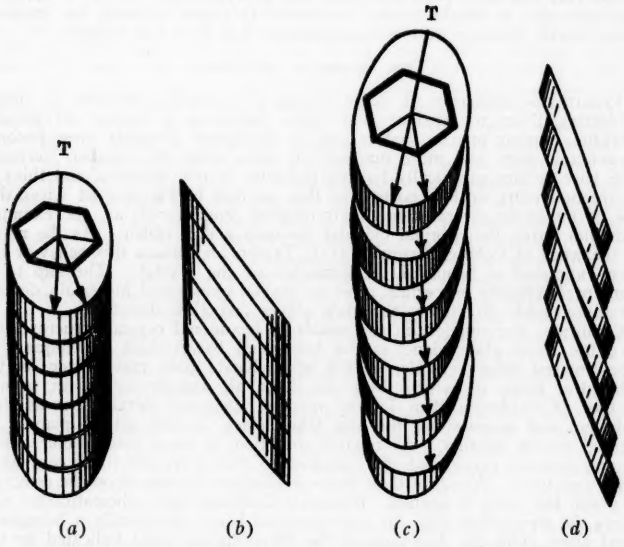

FIGURE 4.—GLIDE PLANE AND GLIDE DIRECTIONS IN A HEXAGONAL CRYSTAL.

tion of these bands, which represent a kind of local avalanches, is a complicated one to which reference will be made later.

If a number of single crystal wires of cadmium, say, are taken, it will be found that the load under which they begin to deform will vary from specimen to specimen. The reason is that what determines the plastic behavior is the shear stress on the glide plane resolved in the glide direction; thus, if a load W be applied to a wire specimen of cross-sectional area A, and the glide plane and glide direction make angles χ and λ with the

[1] L. C. Tsien and Y. S. Chow, Proc. Roy. Soc. A, 163, 19, 1937.
[2] E. N. da C. Andrade and Y. S. Chow, Proc. Roy. Soc. A, 175, 290, 1940.
[3] L. C. Tsien and Y. S. Chow, loc. cit.
[4] Mercury can be prepared in greater purity than any other metal.
[5] E. N. da C. Andrade and R. Roscoe, Proc. Phys. Soc., 49, 152, 1937.
[6] K. M. Greenland, Proc. Roy. Soc. A, 163, 28, 1937.

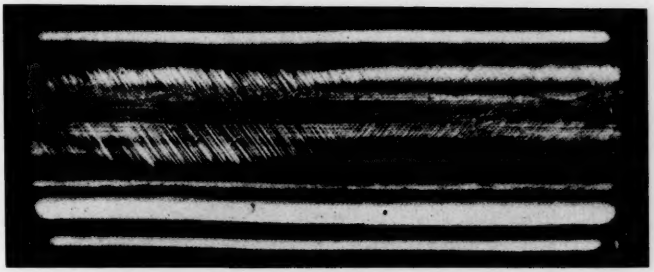

FIGURE 9.—SLIP BANDS OF PURE MERCURY.

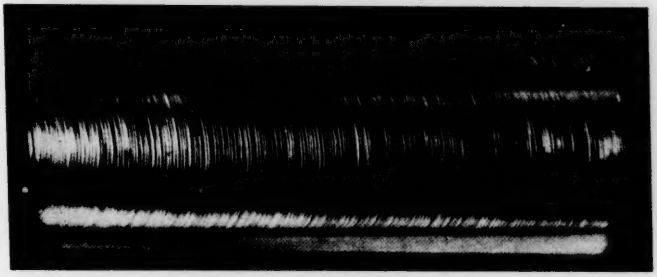

FIGURE 10.—SLIP BANDS OF PURE MERCURY.

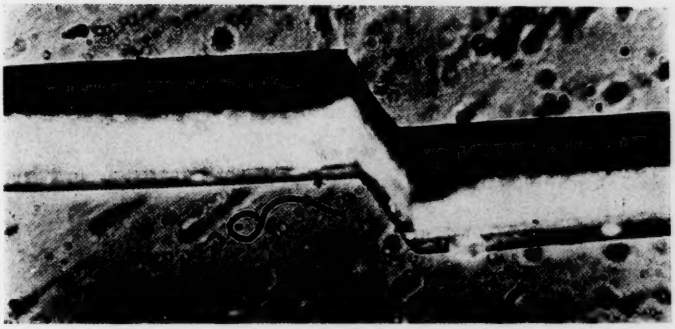

FIGURE 11—SLIP IN CADMIUM CRYSTAL WITH PLANES NEARLY NORMAL TO AXIS.

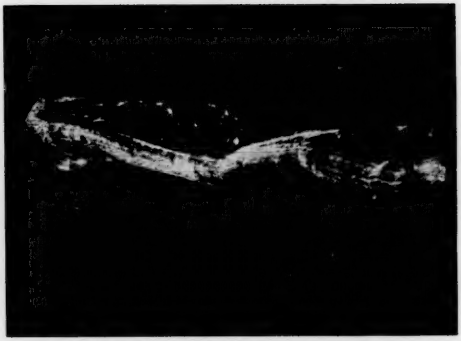

FIGURE 12.—DOUBLE GLIDE IN CADMIUM: TWO DIRECTIONS EQUALLY FAVORABLE.

FIGURE 13.—DOUBLE GLIDE IN MOLYBDENUM AT 300 DEGREES C.: TWO CRYSTALS SIDE BY SIDE.

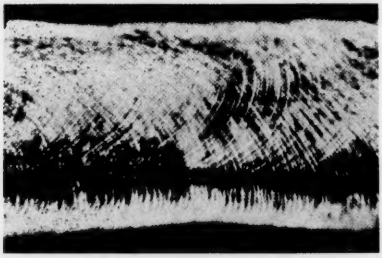

FIGURE 14.—DOUBLE GLIDE IN MERCURY: TWO PLANES EQUALLY FAVORABLE.

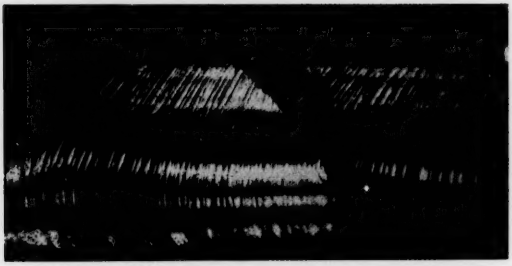

FIGURE 15.—TWINNING IN A MERCURY CRYSTAL.

axis of the wire, then the so-called critical shear stress, which is a constant for the metal, is given by

$$S = W/A \cos \lambda \sin \chi$$

The behavior is complicated by the fact that as the wire is stretched the planes rotate: they tend to move all parallel to one another, and can only do this by bending near the grips. The geometry of glide has been fully worked out, and explains some of the extraordinary features which single crystals show under stress. For instance, if the glide planes of cadmium are nearly normal to the axis of the wire, glide may take place at one or two places only, giving the strange appearance shown in Figure 11[1], which I call "stove-pipe slip." The main part of the wire bends very slightly so as to keep the mid parts of the short elbows in line with the direction of tension. Another strange appearance is shown in Figure 12[2]; this takes place when two of the glide directions are equally favorable, so that short lengths of the crystal slip alternately in one and the other direction. Double glide in cubic crystals and other crystals where there is more than one set of equivalent glide planes can also take place on different sets of glide planes, which either happen to be equally favorably disposed initially, or become so in consequence of changes in the course of extension. Figure 13[3] shows a curious case, where two crystals have grown side by side in a molybdenum wire, and on extension at 300 degrees C. double glide has taken place in each. Figure 14[4] shows double glide in a mercury crystal. Double glide on two sets of planes, disposed equally favorably, can be also seen in Figure 9.

After a certain amount of extension has taken place the crystal may show another phenomenon, that of twinning, the molecules moving so that the crystal structure in one part of the twin is the mirror image of that in the other part of the twin. Figure 15[5] shows a boundary where twinning has taken place in a mercury crystal. The reason for this twinning is still obscure, but it adds to the difficulties of the subject. In cubic and rhombohedral metals, where there are groups of exactly equivalent glide planes, on which glide may take place simultaneously or consecutively, things may be very complicated. In a general way, however, the geometry of the glide in single crystals has been worked out and most of the broad phenomena can be explained. If it is remembered that the individual crystallites in a polycrystalline metal can behave in any of the ways indicated, and that crystallites differently disposed behave in different ways in one specimen, the complexity of the problems of an ordinary metal can easily be realized.

Strength of Metal Single Crystals.

When we turn to the question of strength we enter a very difficult field. The simplest case is that of brittle fracture. The determining factor here is the tension normal to the plane of rupture. If the critical tension N which leads to fracture; the critical shear stress S; and the crystal structure are such that a crystal can be so disposed that the stress resolved normal to the plane of rupture is N while the stress resolved along the glide direction is less than S, fracture without glide can take place. With crystals of hexagonal structure it is possible to realize these conditions, and we find the recorded critical tensions run from 120 to 18 bars with different metals and different temperatures, for instance, 20 bars for the basal plane of zinc at —185 degrees C. This compares with 1.3 kilobar for polycrystalline zinc and a theoretical value of some 100 kilobars.

[1] K. M. Greenland, loc. cit.
[2] E. N. da C. Andrade and R. Roscoe, loc. cit.
[3] E. N. da C. Andrade and R. Roscoe, loc cit.
[4] E. N. da C. Andrade, unpublished.
[5] E. N. da C. Andrade and P. J. Hutchings, Proc. Roy. Soc. A, 148, 120, 1935.

Brittle fracture is, however, rather the exception than the rule. For metals of cubic structure, where there are many alternative equivalent glide planes, the values of N and S and the geometry of the crystal apparently prohibit it. Even with hexagonal metals glide is the normal occurrence.

The inception of glide, which takes place by sudden avalanche-like slipping in the neighborhood of isolated planes, occurs at very low stresses, of the order of 20 bars, much less than one-thousandth of the theoretical strength. If we wish to consider rupture, we are confronted with a variety of processes that can take place before actual fracture. We have in general a progressive hardening as movement proceeds, but quite often, especially with hexagonal metals, we ultimately get twinning, leading to glide on a fresh set of planes at a new inclination, which is followed by fracture. In the case of hexagonal metals, where brittle fracture can be brought about by suitable geometrical conditions, the glide which takes place before rupture can either raise or lower the breaking strength, according to temperature. For these metals the breaking strength is, however, of the same order as that in the case of brittle fracture. In the case of certain metals which will be mentioned later glide can lead to very great hardening, so that stresses of nearly a hundred times the critical shear stress can be supported, but here, as will be pointed out, the result of glide is that the specimen has lost much of its monocrystalline character. It remains true that a single crystal is very soft.

Before we look at any further facts, it may be well to consider the attempts that have been made to explain theoretically the outstanding difficulties offered by the softness of the crystals—their proneness to glide under very small shear stresses. The atoms are bound in their places by interatomic forces, the potential of the field of force having a minimum at the position where the atom is in equilibrium. If an atom is moved from this position the force tends to restore it; the case is similar to that of a table covered with a regular network of depressions, with a ball lying in each depression. Now consider the force required to move all the balls at once in a given direction, so that each rises out of its hole and falls into the next hole in that direction. It will clearly be the force required to move a single ball multiplied by the number of balls, and this is equally true when a ball lies in every hole and when a few vacant holes are included in the array. The low value of the critical shear stress cannot be accounted for on the basis of the simultaneous movement of all atoms in a glide plane.

The position may perhaps be clarified by considering simply a row of cylinders touching one another, with a second row of cylinders resting on them, as in a model which I have constructed (Figure 16). The force required to move one whole row over the other is considerable. Suppose, however, that the number of cylinders in the upper row is one or two fewer than that in the lower row. A place of non-fit, or dislocation, indicated by a white line above it, is arranged at one end of the row, towards the right, there being four in the upper row to five in the lower row here. To make the model resemble the atomic case all the upper cylinders in the model are connected by a long strip of elastic band. If we now run our hand over the upper row, from right to left, which is equivalent to imposing a force from left to right on the lower row, it is easy to make this dislocation travel along: at any given moment we are only moving a few cylinders through the periodic field of gravitational potential. After the dislocation has traveled, however, every cylinder has moved one place and the whole row is displaced relatively to that below, although at any given movement we were only moving a few cylinders.

The fact that glide takes place at shear stresses so far below the theoretical value shows that a whole sheet of atoms cannot travel at once over a neighboring sheet. The conception of the glide mechanism, which we owe to Polanyi, G. I. Taylor, and Orowan, postulates that in the metal crystal there are small localities in which the crystal is out of joint, places of misfit

something like that represented very crudely in our model. Under shear the whole crystal plane does not move by one atomic spacing simultaneously: rather the misfit runs easily through the crystal, leaving in its wake atoms properly spaced. In a sense, then, the weakness of the crystal is due to imperfections which are somehow inherent in the structure. A perfect crystal might have theoretical strength, but probably no such thing exists.

There is considerable doubt and diversity of opinion as to the origin and end of the dislocations. G. I. Taylor[1], for instance, when proposing a very suggestive but rather formal two-dimensional model considers that the stress produces in the metal crystal a multitude of regularly spaced dislocations, which run a certain distance and then end their course in flaws which he postulates. It is difficult to see on this picture anything to prevent the effect being reversible, which it certainly is not. Orowan considers that the dislocations are created at microscopic "Griffiths cracks," sharp notches where the stress has values considerably above the average, and that they run right across the specimen. W. G. Burgers and Kochendörfer believe that the crystal is built up of small blocks in which the orientation of the crystal structure differs slightly from block to block, for which there is X-ray evidence with rock salt and some other crystals. The dislocations are then supposed to start from the boundary of the blocks: how they are stopped is not clear. I have not time to discuss the details and difficulties: the general conception of a travelling misfit seems, however, to be undoubtedly the proper basis for any theory of glide.

Another fruitful conception is that of cracks, notches, or other regions of local peculiarity leading to increased local stress. Thinking of a wooden plant we may, perhaps, be allowed to call these hypothetical regions "knots." If these regions of stress concentration are small the stress will depend upon the irregularities of atomic motion in the neighborhood—the so-called temperature fluctuations, which are the more violent the higher the temperature. Every now and then the atoms so will dispose themselves that, speaking loosely, the crack is sharper: the longer one waits the greater the chance that the local stress reaches some high critical value. This conception is due to Becker[2]: it has been elaborated by Orowan. The rate of flow of a crystal under stress will be proportional to the frequency with which the critical stress is reached at the local "knots." This leads to the following formula for the rate of glide, u:

$$u = Ce^{-V(R-qS)^2/2GkT}$$

where R is the true critical shear stress, S the average externally-applied stress, q a factor to represent the increase at a "knot," G the shear modulus, k Boltzmann's constant, T the absolute temperature, and V a certain small volume, full consideration of which would take us too far.

This formula has certain interesting consequences. It shows the rate as depending very strongly on the stress; according to it, flow should take place at any stress, but if the stress is below a certain value the flow is unobservably small. This is the kind of thing we must expect to find in theories based upon probabilities.

If this view is correct, we cannot define a critical shear stress for a single crystal by saying it is the resolved shear stress at which glide begins to take place, for this will depend upon what rate we are prepared to consider noticeable. I have suggested that we define it as that at which a rate of 1 per cent per hour takes place. This seems a convenient arbitrary figure, but it is nothing more. The exact rate does not make much difference: in the case of very pure cadmium, considered by Roscoe and myself[3],

[1] Proc. Roy. Soc. A, 145, 362, 1934.
[2] Phys. Z., 26, 919, 1925.
[3] Proc. Phys. Soc., 49, 152, 1937.

stresses of 1.5, 1.3, and 1.0 bar gave rates of 1 per cent per 13 minutes, per hour, and per 7 hours respectively. A stress of 0.5 bar still gave a flow, but of only 1 per cent in 13.5 days. In the neighborhood of the suggested rate, with cadmium, for instance, the rate is doubled by about an 8-per cent increase in the stress.

This way of regarding critical shear stress as the stress at which a determined slow rate of glide begins also explains a remarkable fact, as pointed out by Orowan, who has emphasized the importance of concentrating on the rate of flow as a property of the single crystal. While the rate of flow at a given stress increases very rapidly with the temperature, at a given stage of the flow, and the strain hardening is also very susceptible to temperature, the critical shear stress is not very much influenced by temperature: even at —235 degrees C. it is, for cadmium, only four times as big as it is a few degrees below the melting-point of the metal. If we take the critical shear stress to correspond to a fixed value of u at all temperatures this means that $(R-qS)^2/T$ is approximately constant, which gives $S = A - B\sqrt{T}$, a relation which agrees quite well with experiment.

The difficulty of interpreting curves showing the shear stress plotted against extension ("hardening" curves) will now be clear. If there is a time element, if flow is taking place all the time, these curves will depend upon the rate at which stress is increased. By working within a region where the rate of flow is very small, and applying the stress fairly rapidly, hardening curves with some meaning can be obtained, but sufficient attention has not always been given to this point. The whole question is a troublesome one, complicated by the effect of recovery, that is, a crystal that has been hardened by strain becomes soft again if left to itself at a sufficiently high temperature: for "soft" metals, like cadmium, room temperature is sufficient. Some workers have attributed the time flow entirely to recovery taking place while the stress is still applied: others have tended to neglect it altogether. Actually true flow in the absence of recovery undoubtedly exists, as can be demonstrated by working at low temperatures, where the rate of recovery must be so exceedingly slow as to be quite negligible: on the other hand recovery undoubtedly plays a part in flow at higher temperatures. Hardening at high temperatures and at low temperatures exhibits different features and requires rather more careful discrimination than has generally been made. The problems of strain hardening can, however, only be glanced at here.

The time element being eliminated as best one can, curves showing the variation of stress with glide[1] at temperatures suitably chosen can be obtained. Some very striking results are available on metals of comparatively high melting-point. Measurements made by Sachs and Weerts[2] and by Osswald[3] show that with a single crystal of nickel a shear strain of 30 per cent produces an increase of shear strength of fourteen times; with copper a shear strain of 67 per cent produces an increase of strength of sixty-eight times; while with silver a shear strain of 95 per cent produces an increase of strength of ninety-two times, all at atmospheric temperature. In many cases the hardening at low temperatures varies as the square root of the glide: at high temperatures it tends to show a linear variation.

Evidence from X-Rays.

We have now to consider how this hardening under strain comes about, and here X-rays furnish a valuable method of attack. If we pass a narrow beam of so-called "white" X-rays, that is, a beam containing a wide con-

[1] By "glide" is meant the shear strain—the relative movement of two parallel glide planes divided by the perpendicular distance between them. The extension is not a suitable parameter to consider, since it will depend upon the inclinations of the glide planes to the axis in the particular specimen.

[2] Z. Physik, 62, 473, 1930.

[3] Z. Physik, 83, 55, 1933.

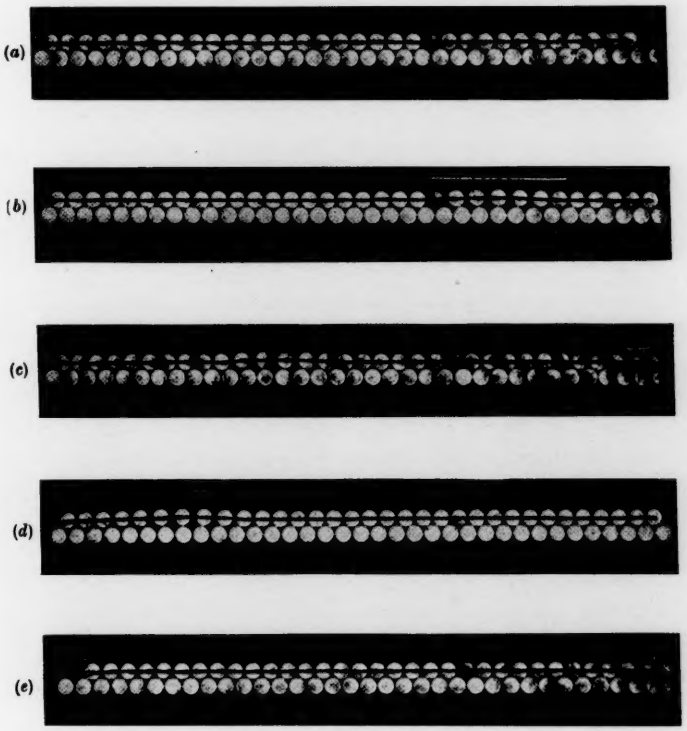

FIGURE 16.—MODEL ILLUSTRATING HOW GLIDE CAN TAKE PLACE BY THE PROPAGATION OF A MISFIT.

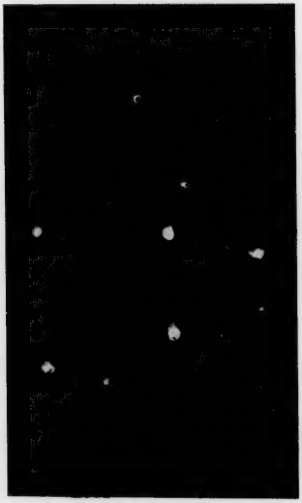

FIGURE 17.—LAUE PHOTOGRAPH OF AN UNSTRAINED SODIUM CRYSTAL.

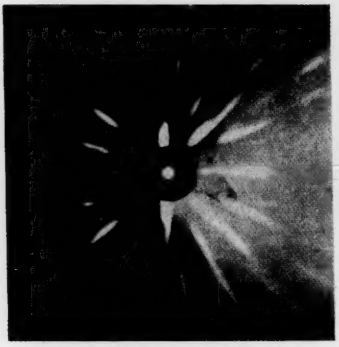

FIGURE 18.—LAUE PHOTOGRAPH OF HEAVILY STRAINED CRYSTAL OF *a*-IRON.

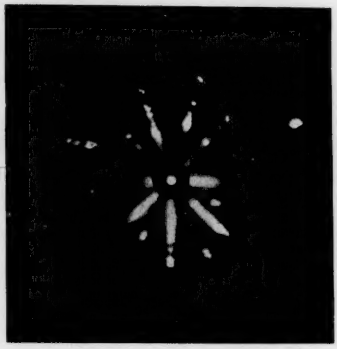

FIGURE 19.—ASTERISMS DUE TO STRAIN IN A SODIUM CRYSTAL.

FIGURE 20.—SLIP IN STRAINED POLYCRYSTALLINE LEAD.

tinuous range of wavelength, through a fixed single crystal, each set of crystal planes will reflect the rays at a determined angle and we obtain, as is well known, a series of spots, the so-called Laue pattern. If the crystal is turned to another position the spots will turn to fresh positions, just as will a narrow beam of light reflected from a mirror if the mirror is rotated, or similarly, if the beam passes through two crystals oriented in a slightly different way, the spots will appear double. Now if we take a Laue photograph with an unstrained single crystal, we obtain a pattern of single spots, as shown in Figure 17[1], obtained with an unstrained crystal of sodium; but if we strain the crystal the spots are drawn out into smears, called "asterisms," just as would be the case in the presence of a large number of crystals, oriented in slightly different ways, so as to give a train of overlapping spots. Figure 18[1], which is a Laue picture of a crystal of α-iron, extended by about 16 per cent, shows such asterisms. The heavy strain has produced a change in the crystal structure; in fact it looks as if the single crystal is strong because it is no longer a single crystal.

With Miss Chow I carried out some experiments[2] on sodium single crystals, extending them by an equal amount at various temperatures and measuring (1) the stress required to produce this extension, (2) the spread of the asterism. It is easy to measure this with sodium, since the asterism takes the form of discrete spots (see Figure 19)[2], due, as we showed, to recrystallization; the separation of the extreme spots represents the range of rotation of crystalline fragments produced by stress. These experiments show that at low temperatures, where the crystal is strong, the spread is much greater than at higher temperatures, where it is weak, for the same strain. We have also been able to establish some numerical regularities. I think, then, that there is now definite evidence that the strengthening is largely due, at any rate, to the break-up of the crystal, to a loss of its single crystal character. Much more experiment is, however, needed to clear up the situation.

Polycrystalline Metals.

This brings us half-way to real metals and the problem which interests the engineer—why are polycrystalline metals strong? There are very few experiments available on the dependence of strength on grain-size. In its simplest aspect this problem demands an absence of plastic deformation, which at once complicates the question, but on the whole it looks as if small grain size is favorable for strength. During plastic deformation slip takes place in the individual crystallites, whose axes are differently oriented, as shown in Figure 20[2], which represents a strained specimen of polycrystalline lead. In some way the boundaries between the crystallites must confer strength, but they cannot do it because they are inherently strong—the number of atoms involved is too small. Expressed in another way the crystal boundaries cannot be regarded as playing the part of the steel rods or expanded metal in reinforced concrete: they are too thin, whatever reasonable strength be attributed to them.

Supposing that, to account for their weakness (low yield-point) as compared with ideal crystals, we accept the hypothesis that the crystallites contain submicroscopic flaws, similar to Griffiths cracks in glass, where the stress-concentration is abnormally high. If a dislocation runs from such a crack it will travel as far as a crystal boundary and will then find, for a region a few atoms thick, at any rate, a place where the regularity necessary for it to propagate itself is lacking—it loses itself, but not because the boundary is hard, rather because it is soft. An experiment which I carried out some time ago may illustrate the point.

[1] E. N. da C. Andrade and Y. S. Chow, loc. cit.
[2] E. N. da C. Andrade, unpublished.

To show the effect of crevasses in weakening a solid, I cut transverse slots in celluloid strips, and compared the strength of these strips with that of other unpierced strips, of equal carrying breadth. The weakening effect of the cuts which was to be anticipated did not, however, show itself. Examination shows that at the ends of the crevasse, where the curvature was greatest and tearing should have taken place, the celluloid had yielded plastically, with slight local deformation made evident by a turbid appearance. The theoretical weakening effect of a crevasse is based upon a postulated rigid behavior of the solid: if the solid yields plastically the local stress is dissipated and the tearing does not take place. The analogy is not a perfect one, but does show how strengthening can take place by weakness. Rather large round holes, cut near the ends of the crevasse, in which the high local stress would lose itself when the crevasse yielded, might also be used to illustrate the point.

There is no doubt that the intercrystalline boundaries play a very important part in the strength of metals. However, the experimental study of the properties of these boundaries is only just beginning, and, although Chalmers[1] has carried out interesting experiments on the subject, much work with different metals and at different temperatures is required before we have enough material to found a comprehensive theory. The properties of single crystals of metals may seem curious and curiously remote from anything that the engineer experiences, but it is in terms of their behavior, and of the behavior of the less regularly ordered atoms in the boundaries between crystals, that we must seek an explanation of the secrets of the ironmaster and of the worker in non-ferrous metals.

Conclusion.

I have tried to indicate to you, then, some of the problems which confront us when we try to explain the simplest features of the mechanical strength of solids, such as the tensile strength, the rupture under shear, work hardening and creep. I have not had time to say much on any of these problems, and if we knew more no doubt my story would have been simpler and more consequential. I have had to lay before you problems which are still in the early stages of attack rather than a compact account of a subject successfully ordered by a comprehensive theory. I would, however, end by a plea that it is a subject well worthy of attention. Even from the industrial point of view the work which I have very tritely and tentatively described is beginning to touch on practice. I have referred to the reinforcing of concrete by glass beams and to hardened glass and glazes, where the state of the surface has had to be considered. I may say also that it has been found necessary to make the tungsten spirals of glow-lamps of single crystal wires, or of wires consisting of very few crystals, for reasons explained by Becker on theoretical grounds, which I have not had time to describe. In a lamp in use the spirals are above the recrystallization temperature, and recrystallization, that is, the transfer of atoms from one lattice to another, is accompanied by marked flow under stress. A single crystal cannot exhibit recrystallization. It is, however, not on these details that I wish to appeal to you.

After the war there will probably be a great movement for the prosecution of research with an immediate practical end in view, and of the importance of such research there is no question, nor will there be any lack of voices shouting in support of its obvious claims. I wish to enlist your sympathy for the type of research which I have been describing, pursued with the more fundamental aim of finding out the mechanism that underlies certain general properties of matter. The work will be undertaken by men

[1] Proc. Roy. Soc. A, 175, 100, 1940.

who long to investigate for learning's sake, lured on by the beauty of the phenomena and the hope of finding a meed of that intellectual, nay spiritual satisfaction which attends the understanding of any small sphere of natural happenings—by men who, varying in ability but not in enthusiasm, desire to be natural philosophers, to use the old and choice phrase. The ends are vast, the approach slow.

The ultimate practical gain may be great, for as we have seen, there are theoretical possibilities of vastly increased strength of materials—where so many of the dreams of inventors are clearly impossible on simple grounds of energy and such-like fundamental considerations, gigantic advances are here, at any rate, not inherently ruled out. Just as the work of J. J. Thomson and his school, carried on without the least thought of practical ends, carried on in the faithful pursuit of scientific truth and beauty, led to the whole modern light electrical industry, so may work of the type which I have glanced at carry in it the seeds of a new metallurgy. I am not suggesting that any of those who have been working in the field are J. J. Thomsons, or even seconds to J. J. Thomson—far from it. I am not appealing to you on the ground of an immediate or ultimate practical gain. I am begging that if the time comes when you are asked whether any shall be allowed to carry on pure experimental research whither the urge leads them, you shall not ask too strictly whether a practical end is in view, but rather, if you think that they are capable and in earnest, are seriously striving in their small way to advance knowledge, that you shall refresh them and wish them God-speed.

NOTE ON METACENTRIC DIAGRAMS.

Emeritus Professor J. J. Welch of Kings College, Newcastle-on-Tyne, is author of the following article taken from the July, 1941, number of The Shipbuilder and Marine Engine-Builder.

This very useful diagram was first introduced by the late Mr. F. K. Barnes of the British Admiralty, and its geometry has received attention from time to time, notably by the late Sir William H. White and Professor P. Jenkins. The first-named, in a paper read before the Institution of Naval Architects in 1878, discussed the variation of shape in the curve of metacenters with varying shape of ship, and, in particular, investigated the condition for the tangent to that curve to be horizontal. He found that this would occur at any particular draught where the center of curvature of the curve of flotation coincided with the position of the corresponding metacenter. For present purposes, the curve of flotation for a particular displacement may be defined as the locus of the center of flotation as the ship is revolved round a longitudinal horizontal axis while maintaining the same displacement; or is the curve to which the several water-lines are tangential. Thus, if at two consecutive points on the curve of flotation normals are drawn to the respective water-planes, they will intersect at some point, O, say, which is the center of curvature of the curve of flotation at that point. In this article, the only center of curvature required is that for the upright position, so that the point O falls on the middle-line of the ship. The formula for obtaining the position of O will be given later.

Professor Jenkins, in his contribution to the Transactions of the same Institution in 1884, deduced formulae expressing the inclination of the tangents to the curve of centers of buoyancy and curve of metacenters in

the metacentric diagram. Thus if ϕ represents the inclination to the horizontal of the tangent to the curve of centers of buoyancy at any point:—

$$\tan\phi = \frac{Ah}{V} \qquad (1)$$

where A is the area of the water-plane, h the depth of the center of buoyancy below the water-line at which the vessel is floating, and V the volume of displacement to that water-line.

Further, if θ is the angle which the tangent at any point on a curve of metacenters makes with the horizontal,

$$\tan\theta = \frac{A}{V}(h + r - BM) \qquad (2)$$

where A, h and V have the same significance as before, r is the radius of curvature of the curve of flotation, and BM is the vertical distance between the center of buoyancy and transverse metacenter.

The value of r is given by Leclert's formula:—

$$r = \frac{dI}{dV} \qquad (3)$$

where $\frac{dI}{dV}$ is the limiting value of the fraction $\frac{\Delta I}{\Delta V}$ (ΔI being the alteration of moment of inertia of water-plane about a longitudinal axis through the center of flotation corresponding to the small increment of displacement ΔV), as ΔV is indefinitely diminished. The alteration of inertia may be positive or negative: in the first case, r is set up above the water-line; in the second place, below it. Leclert's formula is valid for all points on the curve of flotation, but, as already indicated, it is only required in this connection for the upright position of the ship.

[Incidentally, it may be mentioned that equation (2) in the foregoing was obtained independently by the present writer and given to his classes in the course of his lectures at Armstrong (now King's) College, Newcastle-on-Tyne. Only in recent months did he discover that he had been anticipated in this by Professor Jenkins.]

Proofs of the formulae quoted above are given in an appendix.

Looking at equation (2), it is seen that, for the tangent to be horizontal, $\tan\theta = 0$, and then, if A is not zero, $r = BM - h$. That is to say, the height of the center of curvature of the curve of flotation above the water-line is equal to the height of M above that line, and the center of curvature thus coincides with M, as Sir William White found was the case. Further, if $h + r$ is greater than BM, $\tan\theta$ is positive and θ, measured anticlockwise, is less than a right angle; if $h + r$ is less than BM, then θ is greater than a right angle.

Comparing equations (1) and (2), it is seen that if for a particular water-line the tangents at corresponding points of the curves of centers of buoyancy and metacenters are to be parallel to each other, r must be equal to BM for that water-line. If, on the other hand, the two tangents are at right angles to each other, θ must be equal to $\frac{\pi}{2} + \phi$, or $\tan\theta = -\cot\phi$. That is, $\frac{A}{V}(h + r - BM) = -\frac{V}{Ah}$, or $BM - h - r = \frac{V^2}{A^2h}$. Thus, for the

tangents to be at right angles, the center of curvature of the curve of flotation must be the distance $\frac{V^2}{A^2h}$ below M.

It may be of interest to examine the diagrams of certain prismatic forms in the light of the foregoing, dealing simply with unit length, since this dimension enters equally into numerator and denominator in the ratios dealt with.

(1) *A Vessel of Constant Rectangular Section of Breadth B.*

Here $r = 0$ throughout since there is no change of inertia with change of draught. $h = \frac{H}{2}$ where H is the draught, and $BM = \frac{1}{12}\frac{B^2}{H}$. Here r can never be equal to BM, and therefore tangents at corresponding points of the curves of centers of buoyancy and metacenters can never be parallel.

$\text{Tan } \phi = \frac{Ah}{V} = \frac{B\frac{H}{2}}{BH} = \frac{1}{2}$, a constant, so that the "curve" of centers of buoyancy is a straight line.

When the tangent to the curve of metacenters is horizontal $h + r - BM = 0$, or $\frac{H}{2} - \frac{B^2}{12H} = 0$, so that $H = \frac{B}{\sqrt{6}} = \frac{B}{2.45}$. In this case since O is in the water-line, the horizontal tangent is the water-line. In many ships of full form, the horizontal tangent falls at or near the load water-line.

For tangents at corresponding points to be at right angles, $BM - h - r = \frac{V^2}{A^2h}$, i.e., $\frac{1}{12}\frac{B^2}{H} - \frac{H}{2} = \frac{B^2H^2}{B^2\frac{H}{2}}$ $(= 2H)$. Therefore, $\frac{B^2}{6} = 5H^2$ or $H = \frac{B}{\sqrt{30}} = \frac{B}{5.48}$.

If any point M on the curve of metacenters is referred to the oblique axes, O_1X_1, O_1Y, as in Figure 1 (O_1X_1 denoting the "curve" of centers of buoyancy), $x = \frac{H}{\cos\phi}$, $y = \frac{B^2}{12H}$, so that $xy = \frac{B^2}{12\cos\phi}$, a constant.

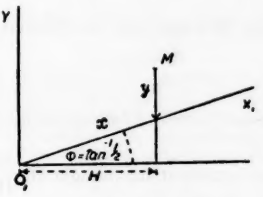

FIGURE 1.

Thus, the curve of metacenters is part of a hyperbola, having O_1X_1, O_1Y, as asymptotes. Figure 2 shows the metacentric diagram for a box-shaped vessel 49 feet wide. For this breadth the tangent to the curve of metacenters will be horizontal when $H = \frac{49}{2.45} = 20$ feet. Further, the tangent to the curve of metacenters will be at right angles to the corresponding tangent to the curve of centers of buoyancy when $H = \frac{49}{5.48} = 8.94$ feet. It is seen from the diagram that these conditions are satisfied.

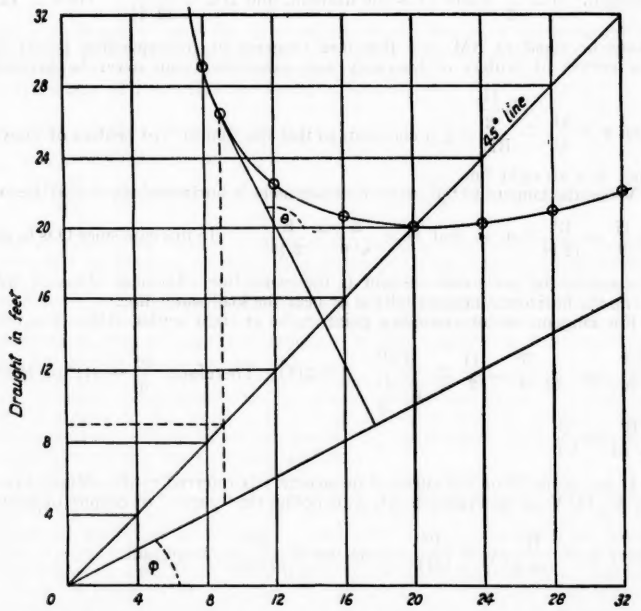

FIGURE 2.—METACENTRIC DIAGRAM FOR BOX-SHAPED VESSEL, DISCUSSED IN SECTION (1).

(2) *Prism of Triangular Section, Floating Vertex Downwards.*

If B (Figure 3) is the breadth at any draught H, $\frac{B}{H} = \frac{B_1}{D} = k$, say a constant, so that $B = kH$.

Then $BM = \frac{\frac{1}{12} B^3}{\frac{1}{2} BH} = \frac{1}{6} \frac{B^2}{H} = \frac{k^2}{6} H$

$I = \frac{1}{12} k^3 H^3$ and $V = \frac{1}{3} k H^2$

$\therefore \frac{dI}{dV} = r = \frac{\frac{1}{4} k^3 H^2}{k H} = \frac{1}{4} k^2 H$ or $\frac{k}{4} B$

Also $h = \frac{H}{3}$. Thus $\tan \phi = \frac{Ah}{V} = \frac{B \times \frac{H}{3}}{\frac{B \times H}{2}} = \frac{2}{3}$ for all draughts. Here, also, the "curve" of centers of buoyancy is a straight line, its inclination to the horizontal being $\tan^{-1} \frac{2}{3}$ or $33\frac{3}{4}$ degrees nearly, and

$$\tan \theta = \frac{A\ (h + r - BM)}{V} = \frac{B\left(\frac{H}{3} + \frac{k^2 H}{4} - \frac{k^2 H}{6}\right)}{\frac{B H}{2}} = \frac{2}{3} + \frac{k^2}{6} \text{ for}$$

all draughts.

Thus the line of metacenters is also straight, radiating from the point of zero draught at a greater angle to the horizontal than the line of centers of buoyancy. It is obvious that tangents at corresponding points of these lines can never be parallel or at right angles, conclusions confirmed by the fact that

r is never $= BM$ nor $BM - h - r = \frac{V^2}{A^2 h}$.

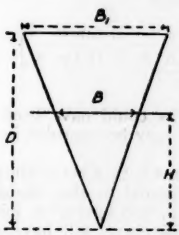

FIGURE 3.

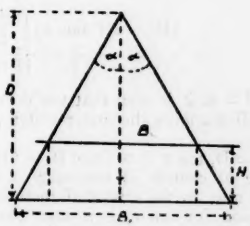

FIGURE 4.

(3) *Prismatic Vessel Similar to the Preceding One, but Floating Vertex Upwards.*

With the information obtained in the preceding investigation, it is easy to obtain from the area and position of centroid of the triangular portion to a particular water-line the area and position of the centroid of the remaining portion, and to calculate its BM. Or the investigation may be done independently as follows:—With the same notation as before, B (Figure 4) $=$ $B_1 - 2H$, tan α, where $\tan \alpha = \frac{B_1}{2D} = \frac{k}{2}$ with the previous notation. The

area below the water-line (and therefore the volume per unit length) $= B_1 H - H^2 \tan \alpha$, and the distance of its centroid below the half-draught level will be given by

$$\frac{H^2 \tan \alpha \times \frac{1}{3}H}{B_1 H - H^2 \tan \alpha} = \frac{1}{6} \frac{H^2 \tan \alpha}{B_1 - H \tan \alpha}.$$

$$\therefore h = \frac{H}{2} + \frac{1}{6} \frac{H^2 \tan \alpha}{B_1 - H \tan \alpha}. \quad \text{Also } V = B_1 H - H^2 \tan \alpha; \; I = \frac{1}{12} (B_1 - 2H \tan \alpha)^3.$$

$$\therefore BM = \frac{1}{12} \frac{(B_1 - 2H \tan \alpha)^3}{B_1 H - H^2 \tan \alpha}.$$

These values enable particulars to be calculated for the metacentric diagram of such a floating body for any given dimensions.

$$\text{Further, } \frac{dI}{dV} = r = \frac{-\frac{1}{2} \tan \alpha \; (B_1 - 2H \tan \alpha)^2}{B_1 - 2H \tan \alpha} = -\frac{\tan \alpha}{2} \times (B_1 - 2H \tan \alpha) = -\frac{B \tan \alpha}{2} = -\tfrac{1}{4} kB.$$

This is, of course, the same numerically as for the same water-line in the previous case, but is negative, indicating that in this case the center of curvature is *below* the water-line. Also,

$$\tan \phi = \frac{Ah}{V} = \frac{(B_1 - 2H \tan \alpha)\left[\frac{H}{2} + \frac{1}{6} \frac{H^2 \tan \alpha}{B_1 - H \tan \alpha}\right]}{B_1 H - H^2 \tan \alpha}$$

$$= \frac{(B_1 - 2H \tan \alpha)\left[\frac{1}{2} + \frac{1}{6} \frac{H \tan \alpha}{B_1 - H \tan \alpha}\right]}{B_1 - H \tan \alpha}$$

Putting $H = 0$, it is seen that $\tan \phi = \frac{1}{2}$, as could have been anticipated, since at small draughts the under-water part may be regarded in the limit as of box form.

When $H = D$, $\tan \phi = 0$, since $B_1 - 2H \tan \alpha$ is then zero. Thus the tangent to the curve of centers of buoyancy is horizontal at that draught.

Referring now to the curve of metacenters, and having in mind the general expression for $\tan \theta$, it is seen that this curve will have a horizontal tangent when $H = D$, since under these conditions A is zero. There will also be a horizontal tangent at the draught where $h + r - BM = 0$; while tangents at corresponding points on the two curves will be at right angles when $BM - h - r = \frac{V^2}{A^2 h}$. Figure 5 shows the metacentric diagram of a prismatic vessel of triangular section, 20 feet deep and 10 feet wide at base, floating vertex upwards. The accompanying table gives particulars of r, h, BM, etc., for several draughts, from which it will be noted that $h + r - BM$ is zero at practically 4 foot draught. The tangent at that draught should therefore be horizontal. as confirmed by the diagram. Also, plotting the values of $BM - h - r$ and $\frac{V^2}{A^2 h}$ on curves to a scale of draught, it is found that they

Draught in ft.	h	r	BM	$h + r - BM$	$\frac{V^2}{A^2h}$
1	.504	—1.19	7.32	—8.006	2.09
2	1.02	—1.13	3.2	—3.31	4.36
4	2.074	—1.0	1.18	—0.106	9.8
8	4.33	—0.75	.281	+3.299	26.3

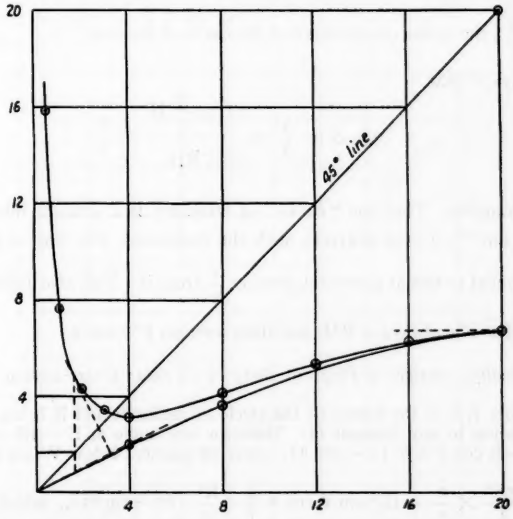

FIGURE 5.—METACENTRIC DIAGRAM FOR PRISMATIC VESSEL OF TRIANGULAR SECTION, VERTEX UPWARDS, DISCUSSED IN SECTION (3).

intersect at 1.75 foot draught, denoting that corresponding tangents are there at right angles—a fact also confirmed by the diagram. Apart from the full-draught position, there can be no other parallel tangents, since nowhere else can $r = BM$.

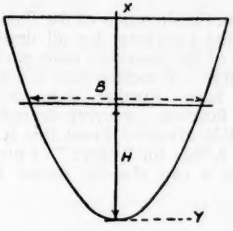

FIGURE 6.

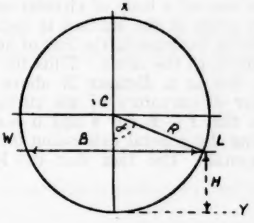

FIGURE 7.

(4). *Prismatic Vessel of Parabolic Cross-section.*

Referring to Fig. 6, we may write $y^2 = lx$, where l is a constant.

Then $\left(\frac{B}{2}\right)^2 = lH$, or $B^2 = 4lH$; $I = \frac{1}{12}B^3 = \frac{8}{3}\,l^{\frac{3}{2}}H^{\frac{3}{2}}$; $V = \frac{2}{3}BH = \frac{4}{3}\,H^{\frac{3}{2}}\,l^{\frac{1}{2}}$; $h = \frac{2}{5}H$; $BM = \frac{I}{V} = \frac{l}{2}$, a constant.

Further, $\frac{dI}{dH} = l\sqrt{l.H}$ and $\frac{dV}{dH} = 2\sqrt{l.H}$

$\therefore \frac{dI}{dV}$, the radius of curvature of the curve of flotation, $= \frac{l}{2}$, a constant, and always = BM.

$$\tan\phi = \frac{Ah}{V} = \frac{B \times \frac{2}{5}H}{\frac{2}{3} \times BH} = \frac{3}{5}$$

for all draughts. Thus the "curve" of buoyancy is a straight line making an angle $\tan^{-1}\frac{3}{5}$ (= 31 degrees) with the horizontal. The line of metacenters is parallel to this at a vertical distance $\frac{l}{2}$ from it. This also follows from the fact that r is always = BM, and therefore $\tan\theta = \tan\phi$.

(5) *Finally, consider a Floating Body of Circular Cross-section.*

In Figure 7, C is the center of the circle of radius R and B is the breadth corresponding to any draught H. Then for any angle α, $B = 2R\sin\alpha$ and $H = R - R\cos\alpha = R(1 - \cos\alpha)$. Area of segment below WL is given by $2\left[\frac{\pi R^2}{2} \times \frac{\alpha}{\pi} - \frac{1}{2}R^2\sin\alpha\cos\alpha\right] = \frac{R^2}{2}(2\alpha - \sin 2\alpha)$, which represents the volume per unit length.

$$BM = \frac{\frac{2}{3}(R\sin\alpha)^3}{\frac{R^2}{2}(2\alpha - \sin 2\alpha)} = \frac{4R}{3} \times \frac{\sin^3\alpha}{2\alpha - \sin 2\alpha}$$

In the case of a body of circular section, the direction of the fluid pressure at every point of the surface is radial, and therefore, for all draughts and all angles of inclination, the line of action of the buoyancy must pass through the center C of the circle. Thus the "curve" of metacenters is a horizontal straight line at a distance R above the base. Further, it is obvious that the center of curvature of the curve of flotation for every draught is also at C, so that $r = R\cos\alpha$ and $h + r = BM$ always. From this it is clear, also, using the general expression for $\tan\theta$, that the "curve" of metacenters is horizontal. The fact that $r = R\cos\alpha$ can also be shown from the

formula $r = \frac{dI}{dV}$. For $\frac{dI}{da} = 2R \cos a \times (R \sin a)^2$

and $\frac{dV}{da} = \frac{R^2}{2}(2 - 2\cos 2a) = R^2(1 - \cos^2 a + \sin^2 a) = 2R^2 \times \sin^2 a$

$\therefore \frac{dI}{dV} = R \cos a.$

From the expression given above for BM, its values for different angles a, and therefore for different draughts, can be obtained and set below the line of metacenters to give the curve of centers of buoyancy. Figure 8 shows the metacentric diagram for R = 10 feet.

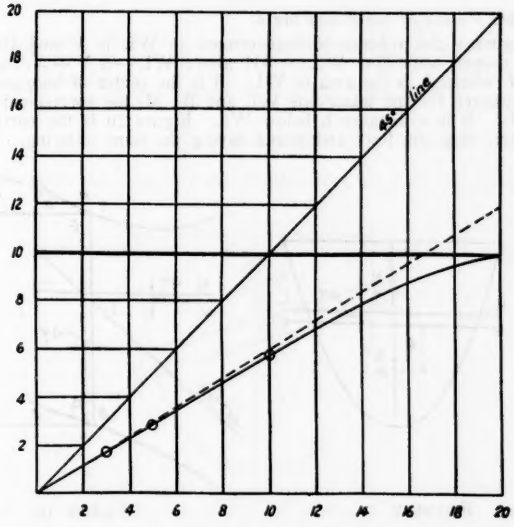

FIGURE 8.—METACENTRIC DIAGRAM FOR VESSEL OF CONSTANT CIRCULAR SECTION, DISCUSSED IN SECTION (5).

From the formula $\tan \phi = \frac{Ah}{V}$ it is seen that when the draught is 2R. A = 0, while the other quantities are not zero. Thus the tangent to the curve of centers of buoyancy will be horizontal at this point. When H =

R, $\tan \phi = \frac{2R \times \frac{4R}{3\pi}}{\frac{\pi R^2}{2}} = \frac{16}{3\pi^2} = \frac{16}{29.61} = .54$

so that $\phi = 28$ deg., about. At zero draught both numerator and denomi-

nator vanish in the expression for tan ϕ and the fraction becomes $\frac{0}{0}$.

But the equation to the circle referred to the axes OX, OY in Figure 7, is $y^2 = 2RH - H^2$, where H, as usual, represents the draught. At limiting small draughts H^2 may be neglected in comparison with H, and therefore at very small draughts the curve may be considered parabolic; so that, in the limit, the tangent will have the same inclination as the parabola, *viz.*, 31 degrees.

It will be noted that the curve of centers of buoyancy is not straight, which is contrary to the statement made on the subject in Sir E. J. Reed's "Stability of Ships" (p. 89).

Appendix

To Find Values of tan ϕ and tan θ.

In Figure 9 the volume of displacement to WL is V and that to the slightly deeper water-line $W_1L_1 - \Delta H$ above WL — is $V + \Delta V$, so that $\Delta V = A.\Delta H$, where A is the area of WL. B is the center of buoyancy and M the metacenter for the water-line WL and B_1, M_1 the corresponding points for W_1L_1. B is a distance h below WL. Figure 10 is the corresponding metacentric diagram, lines and points having the same lettering. Thus, the

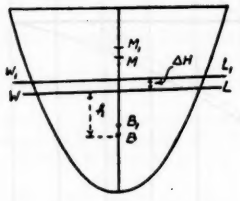

Figure 9.

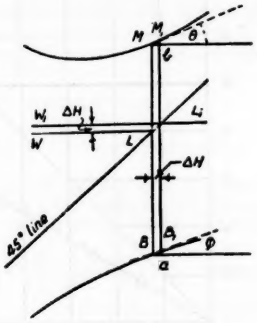

Figure 10.

distances aB_1, bM_1 of Figure 10 are the same as BB_1, MM_1 of Figure 9. Considering the first figure and taking moments about B

$$(V + \Delta V)\, BB_1 = \Delta V \left(h + \frac{\Delta H}{2}\right) = A.\Delta H \left(h + \frac{\Delta H}{2}\right)$$

$$\therefore \frac{BB_1}{\Delta H} = \frac{A\left(h + \frac{\Delta H}{2}\right)}{V + A.\Delta H}$$

and thus, in the limit, when ΔH is indefinitely diminished,

$\frac{BB_1}{\Delta H} = \frac{Ah}{V}$. But $aB_1 = BB_1$ and $\frac{aB_1}{\Delta H}$ is, in the limit, tan ϕ.

$\therefore$ tan $\phi = \frac{Ah}{V}$.

Further, $bM_1 = B_1M_1 + aB_1 - BM$, so that if I is the moment of inertia of WL about its middle-line and $I + \Delta I$ that of W_1L_1,

$$bM_1 = \frac{I + \Delta I}{V + \Delta V} + \Delta H.\ \tan\phi - \frac{I}{V} = \frac{V.\Delta I - I.\Delta V}{V(V + \Delta V)} + \Delta H.\ \frac{Ah}{V} =$$

$$\frac{\Delta I}{\Delta V}.\ \frac{\Delta V}{V + \Delta V} - BM.\ \frac{\Delta V}{V + \Delta V} + \frac{A.\Delta H.h}{V}$$

where $\Delta V = A.\Delta H$ $\therefore bM_1 = A.\Delta H\left[\frac{\Delta I}{\Delta V}.\ \frac{1}{V + \Delta V} - \frac{BM}{V + \Delta V} + \frac{h}{V}\right]$

$$\therefore \frac{bM_1}{\Delta H} = A\left[\frac{\Delta I}{\Delta V}.\ \frac{1}{V + \Delta V} - \frac{BM}{V + \Delta V} + \frac{h}{V}\right].$$

Now, in the limit, when ΔV is indefinitely diminished $\frac{\Delta I}{\Delta V}$ becomes $\frac{dI}{dV} = r$ (see below) and $\frac{bM_1}{\Delta H}$ is the value of $\tan\theta$ at M. Thus $\tan\theta =$

$$\frac{A}{V}\ (r + h - BM).$$

To Determine the Value of the Radius of Curvature of the Curve of Flotation.

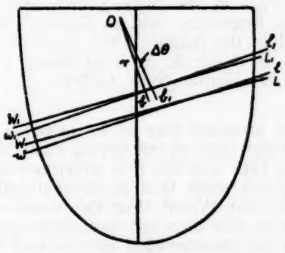

FIGURE 11.

Let WL in Figure 11 be a water-line at any angle θ corresponding to a volume of displacement V, and W_1L_1 a parallel water-line with the small extra displacement ΔV. Also let wl be a water-line inclined at a slight additional angle $\Delta\theta$ also cutting off the volume of displacement V. For this to be the case, WL and wl must intersect at the center of flotation of WL, the volumes of wedges on opposite sides being equal. In the same way, let w_1l_1 be parallel to wl, the displacement to w_1l_1 being the same as to W_1L_1. If b is the center of buoyancy of the layer between WL and W_1L_1, in the limit, when the thickness of the layer is indefinitely diminished, b becomes the center of flotation of the water-line WL; similarly b_1, the center of buoyancy of the layer between wl and w_1l_1 becomes in the limit the center of flotation of the water-line wl. Thus b and b_1 are, in the limit, consecutive points on the curve of flotation; and if normals are drawn from these points to their respective water-lines, they will intersect at O, which in the limit becomes the center of curvature of the curve of flotation and ob becomes r, the radius of curva-

ture of that curve where WL touches the curve. It is seen that $bb_1 = r.\Delta\theta$ where $\Delta\theta$ is in radians. Now the volume of the layer between wl and w_1l_1 differs in shape from that between WL and W_1L_1 only in the fact that a wedge between the upper pair of lines W_1L_1 and w_1l_1 has been moved from left to right, and a wedge between the lower pair has been transferred from right to left. Thus, the movement of the center of buoyancy from b to b_1 depends on the net moment of transfer of these wedges divided by the volume ΔV of the layer. If I is the moment of inertia of WL about a longitudinal axis through its center of flotation, the volume moment of transfer of its wedges is given by $I.\Delta\theta$. Similarly, the moment of transfer of the upper pair of wedges is given by $(I + \Delta I)\Delta\theta$, where $I + \Delta I$ is the moment of inertia of

W_1L_1, $\therefore bb_1 = \dfrac{(I + \Delta I)\Delta\theta - I.\Delta\theta}{\Delta V} = \dfrac{\Delta I}{\Delta V}.\Delta\theta$. But $bb_1 = r.\Delta\theta$; $\therefore r = \dfrac{\Delta I}{\Delta V}$ which in the limit becomes $r = \dfrac{dI}{dV}$. This is true for all points on the curve of flotation.

COLLOIDAL FUEL.

The following paper on the technical problems involved in economic use of colloidal fuel was read at the Semi-Annual Meeting of the American Society of Mechanical Engineers, held in Kansas City last June, and was subsequently published in the August, 1941, issue of Mechanical Engineering. The author, J. E. Hedrick, is a professor in the Department of Chemical Engineering, at Kansas State College.

In 1879 (6)* it was proposed that oil be mixed with pulverized coal in order to get a fuel cheaper than oil but having many of its advantages. The work of Bates (1) in 1918 was the first important attempt at commercial utilization. An effort was made then to develop colloidal fuel for use in submarines. After the first World War this work was dropped. During the last decade efforts in this direction have been renewed in Europe, but there has been almost no consideration of this fuel in the United States. Reports of the work already done are sketchy and cover a limited number of tests. The review by Manning and Taylor (4), however, is a most complete account of previous research.

Presumably the favorable features of colloidal fuel might be listed as follows:

1. It may be cheaper than oil.
2. It may be handled like oil in pipes and pumps.
3. Its burning efficiency should equal or exceed that of oil.
4. It may enable better utilization of storage space.
5. It may permit conservation of our petroleum, or at least release more of it for more profitable markets.
6. Its use would eliminate the danger of spontaneous combustion now present in coal storage.
7. The ash and moisture content would be less than that of the coal and the heat value on a volume basis higher than for either oil or coal.

* Numbers in parentheses refer to Bibliography at end of paper.

8. Its use should permit better utilization of solid fuels such as low-rank bituminous, lignite, and cannel coals.

The most promising uses for colloidal fuel would naturally be those in which advantage could be taken of one or more of the features just mentioned. Outstanding examples are railroad, Diesel, and marine fuels, and in stationary power plants now using pulverized coal. It seems probable too that these oil-coal mixtures may have other important commercial uses than as fuel. Their use as a chemical raw material shows definite promise. The development of such a fuel does present technical problems, however, such as stability, flow through pipes, atomizing, grinding the coal, and mixing the coal and oil.

Stability.

A practical requirement for the satisfactory use of oil-coal fuels is that the coal remain in suspension. The degree of stability is dependent on many variables, the most important of which are size of the coal particles, viscosity of the oil, density of the coal and oil, and the electrostatic charges on the particles. When settling occurs the coal packs tightly in the bottom of fuel tanks and other equipment. This settling does not follow Stokes's Law. The coal particles are of random size and shape, and they interfere with one another.

Stability may be produced by the addition of a third substance to the mixture. Many such addition agents can be used, but it is now recognized that there are three distinct types. These are emulsifiers, gel formers, and surface-active agents. With cracked residue fuel oils, because of their higher specific gravity and viscosity, no stabilizing agent is necessary unless the fuel is heated to temperatures higher than 120 F. Thus there is no stability problem in storage of the mixtures, but only in handling them where they must be heated before pumping and burning.

The preparation of ores by oil flotation is an excellent example of the use of an emulsion to produce a stable suspension. Experiments have been carried out on the stabilization of coal particles in oil using water-in-oil emulsions. The particles are held in suspension by being entrained at the water-oil interface. The use of emulsions, however, is entirely impracticable. They are easily broken by heating or mechanical agitation and they produce stability only with very small amounts of coal. Furthermore, the presence of the water would lower the heating value of the fuel.

Gels are produced by adding rubber, metallic soaps, and various fats and waxes to the oil. The function of these agents is to increase the viscosity. The trivalent aluminum soaps are more effective than the soaps of bivalent calcium, zinc, lead, and magnesium. From 1 to 5 per cent of these soaps, based on the weight of the finished fuel, must be added to prepare stable suspensions. The use of soaps would be expensive and the increase in viscosity would make pumping more difficult. The gels are sensitive to temperature change. Due to the "stringiness" they impart to the fuel, satisfactory atomizing at the burner tip is impossible.

When 0.05 to 0.2 per cent by weight of the condensation product of a fatty acid is added to an oil-coal mixture, a stable suspension is produced. This is an agent of the surface active type. The mechanism of stabilization is not clearly understood, but it is known that a surface-active agent is selectively adsorbed on the surface of the coal particles and causes a lowering of the interfacial tension between the coal and the oil. Furthermore, it causes greater uniformity in the electrostatic charges on the coal particles.

The charges on coal particles are presumably caused by the friction during grinding. Since coal is not a homogeneous material, both positive and negative charges exist, even on different areas of the same coal particles.

The oil is such a poor conductor that these charges are neutralized slowly. This means that the agglomeration of the particles with resultant settling proceeds slowly but definitely. The addition of the surface-active agent produces negative charges on all the particles. This effect is shown by comparison of the pictures of the two suspensions in Figure 1.

Surface-active agents do not increase the viscosity of the suspensions appreciably. They are effective over a wide temperature range and they can be used with any petroleum fraction from gasoline to the heaviest fuel oil. A study of the economics of colloidal fuel (3) limits stabilizer cost to approximately 50 cents per ton. It is believed that this cost can be met with a surface-active material, because of the small amount required.

Experiment has shown that coals of a size larger than 75 per cent through 200 mesh are not satisfactory for making colloidal fuels. Larger sizes cannot be successfully stabilized. On the other hand, smaller sizes require much less stabilizer. Coal of about 98 per cent through 200 mesh seems to be most desirable. Finer coal would be too costly to grind. These results are based on two samples of washed bituminous coal of 1.32 and 1.26 specific gravity, respectively. Probably the results would be equally applicable with other similar coals.

Flow and Atomizing.

No property of oil-coal mixtures is more important than the viscosity. The oils for the most part from which colloidal fuels can be made are viscous and the addition of coal to them increases the viscosity. This property is closely connected with the resistance of flow of these fuels in pipes and with the spraying from burner tips for efficient combustion.

Oil-coal mixtures were prepared using six oils of widely varying gravities and viscosities. The physical properties of these oils are shown in Table 1.

Table 1.—Physical Properties of Oils Used in Experimental Work

	A. P. I. gravity, 60 F.	Furol viscosity, 122 F.
(1) Virgin gas oil	34.1	5.3[a]
(2) Cracked gas oil	29.5	7.3[a]
(3) Cracked residue fuel oil	8.5	18.5
(4) Cracked residue fuel oil	12.0	54.7
(5) Cracked residue fuel oil	10.9	112.7
(6) Cracked residue fuel oil	4.8	318.0

[a] Extrapolated values.

The effect of the coal concentration (weight per cent) on the viscosity of these mixtures at 210 F. is shown in Figure 2. The steep slope of the curves at the higher coal concentrations should be emphasized. When a mixture contains more than 50 per cent coal it becomes a thick paste, which would not flow through pipes. There is a possibility, however, that such fuel pastes might be handled with a screw conveyor or some type of mechanical pusher. In general, the shape of these viscosity-concentration curves is the same at other temperatures.

Because of lack of information, a study has been made of the friction in pipes during the flow of these fuels. The equipment is shown diagrammatically in Figure 3. The fuel was allowed to flow from a tank through a test line 22 feet long into a weighing tank. A constant head of 5 feet was maintained by means of an overflow pipe in the upper tank. A by-pass on the discharge end enabled the fuel to be recirculated and the overflow was also pumped back. Test lines from ½-inch to 3 inches could be used inter-

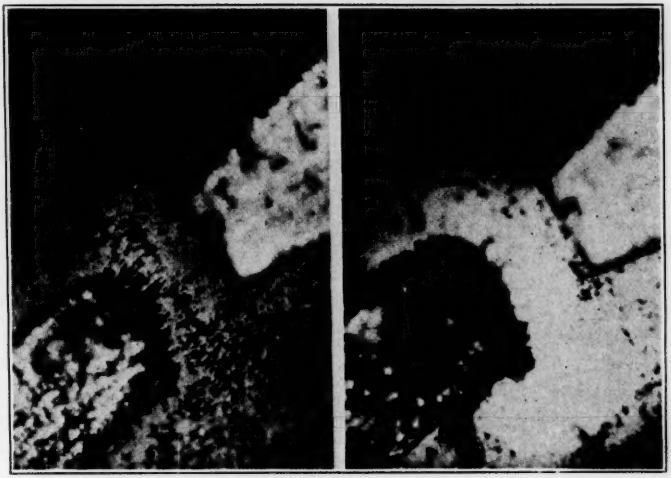

FIGURE 1.—ELECTROSTATIC CHARGE ON COAL PARTICLES IN OIL.
(Left, no stabilizer; right, 0.2 per cent of surface-active agent.)

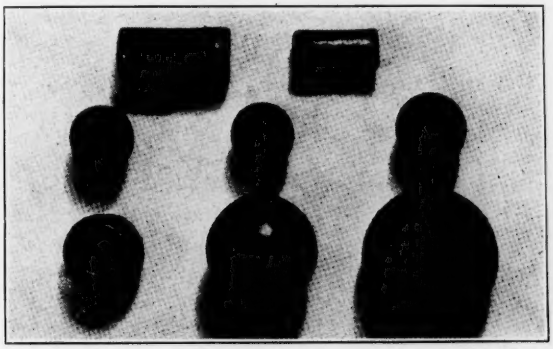

FIGURE 7.—BALLS AND RODS USED FOR COAL GRINDING.

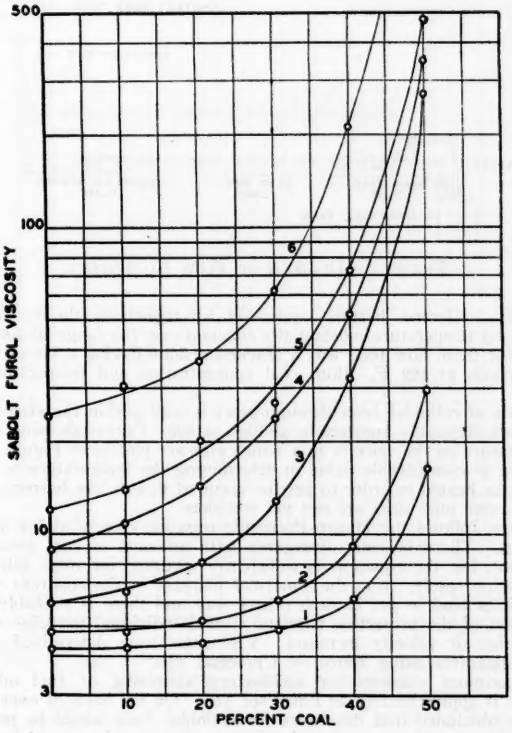

FIGURE 2.—EFFECT OF COAL CONCENTRATION ON VISCOSITY AT 210 F.

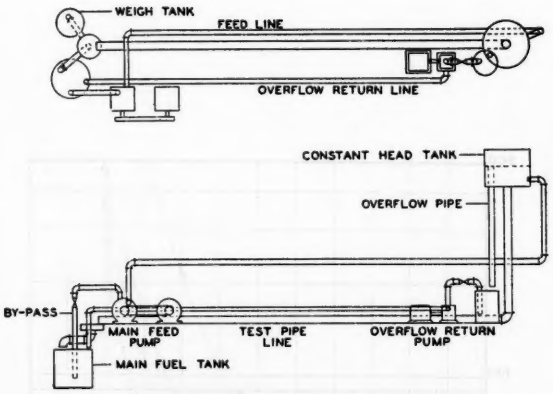

FIGURE 3.—DIAGRAM OF FLOW EQUIPMENT.

changeably. Electric heaters controlled by adjustable thermostats could maintain any temperature substantially constant over the range 80-200 F. The oil used for these flow tests was a cracked residue having a viscosity of 203 Furol seconds at 122 F. Both coal concentration and temperatures were varied.

The flow of colloidal fuels through pipes is well within the viscous range. Seldom will Reynolds' numbers be as high as 500. Curves showing the effect of temperature on the velocity in a 1-inch pipe are plotted in Figure 4. Such curves are of considerable value in determining the temperature to which a fuel must be heated in order to get the required flow to the burner. Similar data for other pipe sizes are not yet complete.

This flow follows the Hagen-Poiseuille equation closely at low coal concentrations. There is some divergence with amounts of coal greater than 30 per cent, but the equation is sufficiently accurate for most calculations. The deviation results from the abnormal increase in the apparent viscosity. The particles tend to get in each others way and there is probably greater deformation of the protective barriers around individual particles as either concentration or velocity increases. Viscosities were determined with the Saybolt apparatus using Furol or Universal tips.

The maximum viscosity for satisfactory atomizing of fuel oil at the burner tip is approximately 40 Furol sec (5). On the basis of experimental tests it is concluded that the figure for colloidal fuels would be nearly the same. With a great many oil-coal mixtures this viscosity would be reached automatically due to the temperature rise caused by the atomizing steam and radiation from the firebox. With others, preheating in the fuel tanks would be necessary.

GRINDING AND MIXING.

The cost of grinding the coal is one of the most important considerations in the economics of colloidal fuel. This cost is a function of so many variables that it is difficult to predict what it would be for a given operation. Roller mills and ball mills are the types most commonly used for pulverized-coal burners. The choice of mill probably lies between the two.

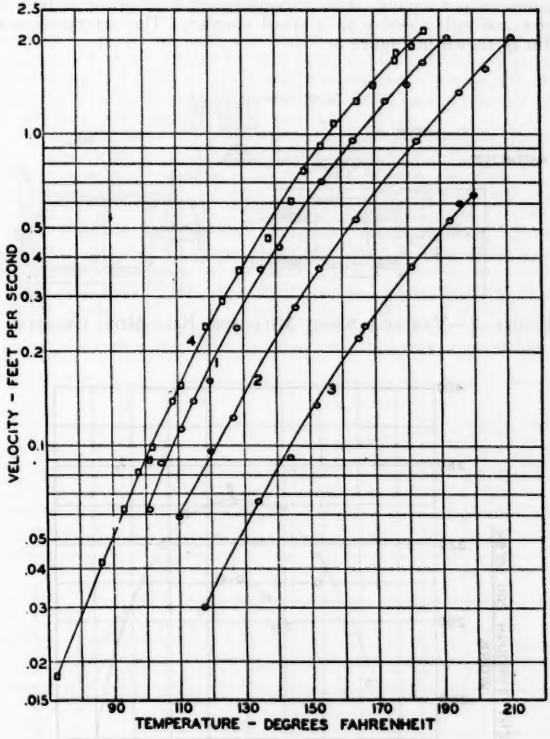

FIGURE 4.—EFFECT OF TEMPERATURE ON FLOW OF FUELS THROUGH 1-INCH PIPE.

(Curve 1, 10 per cent coal; Curve 2, 20 per cent coal; Curve 3, 30 per cent coal; Curve 4, fuel oil.)

In this work, studies have so far been confined to ball-mill grinding. The mill was that used for the determination of A. S. T. M. ball-mill grindability. It was driven by a ½-horsepower motor and equipped with a system of variable-speed pulleys. By mounting the motor on an adjustable platform the speed of the mill could be closely controlled up to speeds of more than 150 revolutions per minute. Power requirement was measured by a suitable wattmeter and mill speed with a speed counter. The arrangement of this apparatus is shown in Figure 5.

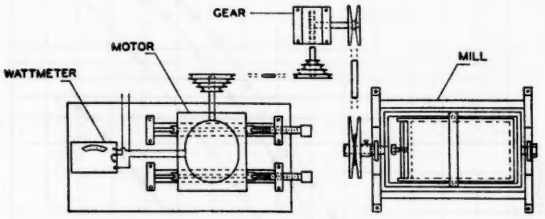

FIGURE 5.—VARIABLE-SPEED MILL FOR BALL-MILL GRINDING.

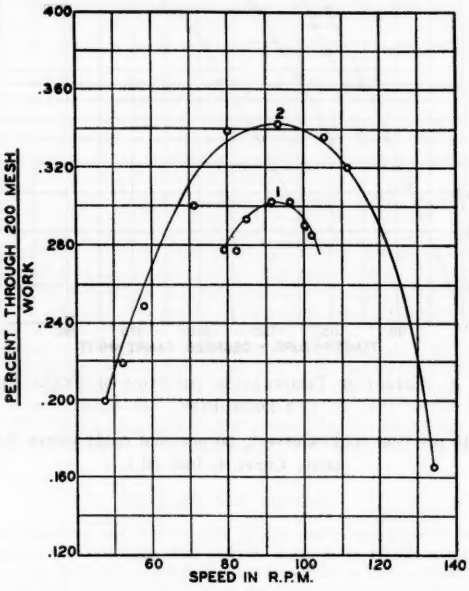

FIGURE 6.—EFFECT OF MILL SPEED ON GRINDING EFFICIENCY.

The speed of the mill has a pronounced effect on the grinding efficiency. Tests were conducted using a charge of 100 1-inch steel balls and 500 g of coal of size—10 and + 200 mesh. The critical speed for the mill was 98 RPM. Critical speed is defined here as the speed at which a ball at the periphery of the mill will just be held in place by centrifugal force. The optimum speed for maximum power input was 92 RPM. Results for two coals are shown in Figure 6. The optimum is therefore about 94 per cent of the critical. Both the optimum and the critical speeds vary with the mill diameter, but their relation to each other is not materially affected. An analysis of Figure 6 leads to the conclusion that the distance the balls fall determines to a large extent the grinding efficiency and therefore that the bulk of grinding is by impact rather than by attrition.

The size and shape of the grinding media materially affect the grinding efficiency. The media used in these tests are shown in Figure 7. Data are far from sufficient to form a general theory, but several observations are interesting. For a required coal size of 90 per cent through 200 mesh, the ¼ × ¾-inch rod gave the highest efficiency. For other sizes different media were more effective. In general, irregular shapes tend to prevent ball coatings which would otherwise lower efficiency. New balls are better than old or worn ones and hollow balls are less effective than solid ones.

No study has been made of grinding aids. Their function is to keep the grinding media free from coatings. Ball coating is influenced by the moisture content of the coal. When the moisture is 2 per cent or less there is no caking, but at 4 per cent moisture this becomes appreciable. Finer

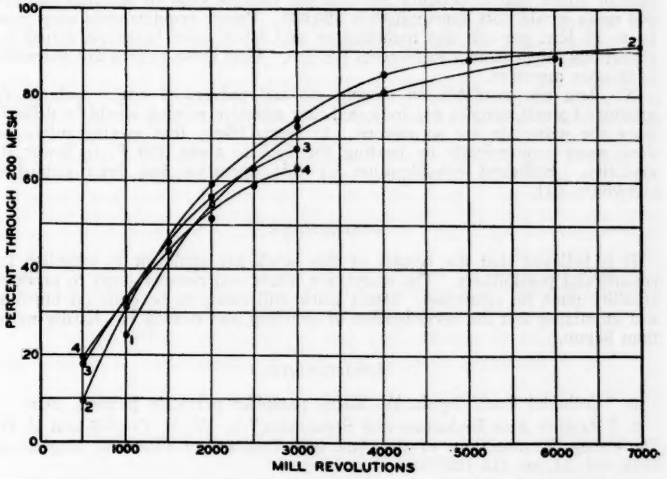

FIGURE 8.—THE EFFECT OF DIFFERENT BALL CHARGES ON GRINDING EFFICIENCY (COAL — 10, + 200).

(Curve 1, 1000 g coal, 30 lb. 9/16-in. steel balls; Curve 2, 800 g coal, 22 lb. 9/16-in. steel balls; Curve 3, 800 g coal, 11 lb. 1-in. steel balls and 11 lb. 9/16-in. steel balls; Curve 4, 800 g coal, 16 lb. 1-in. steel balls and 6 lb. 9/16 in. steel balls).

coal, of course, cakes more readily. In most cases the media and grinding conditions can be adjusted so as to prevent the formation of ball coatings.

Mixed charges of media influence the amount of product of a definite size. The results with various ball charges are shown in Figure 8. These data emphasize that greater amounts of a smaller size are produced by using smaller balls. However, this is true only when the feed is proportionately fine. A definite size of grinding media grinds most efficiently within a certain size range. Thus, mixed charges are often a distinct advantage, each size operating most efficiently on a different size of coal.

A report on the ball-mill grinding of ores by Coghill and De Vaney (2) contains much information of use. The capacity varies as the 2.6 power of the diameter and nearly directly with the length. Increasing the circulating load increases the efficiency and capacity of a definite size. Lighter feeding is better and the load should be just sufficient to fill the interstices of the balls at rest.

Much work must yet be done on coal grinding. The information available is much too general for accurate use. Feed rates, mill shapes, and classification have hardly been investigated at all. An interesting possibility, also, is that of grinding the coal in oil. The coal might be ground first in a disk mill to 16 mesh or finer and then fed with the oil into a ball mill. Thus the fine grinding and mixing could be done in one operation. On the other hand, wet grinding would introduce complications in the way of fineness control, variations in dilution, and gumming up of the equipment.

A few cost figures for estimating purposes only are available. A ball mill or roller mill to produce 2-3 tons per hour of coal 90 per cent through 200 mesh would cost approximately $20,000. Power requirements vary from 10 to 35 Kw. per ton and maintenance and labor costs based on actual installations range from 3 to 10 cents per ton. That these figures are indefinite is at once apparent.

No data are available on mixing oil and pulverized coal. Laboratory mixing of small samples has indicated that effective mixing would be difficult since the materials are so viscous. It seems likely that mixing might be done more economically by heating the fuel to about 200 F. to lower its viscosity. Sufficient consideration of this operation has been published elsewhere (3).

Conclusions.

It is believed that the results of this work are sufficient to establish its commercial possibilities. The search for better and cheaper ways to produce stability must be continued. Much work still needs to be done on burning and atomizing and the investigation of grinding and mixing has hardly more than begun.

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CENTRIFUGAL GOVERNORS.

Credit for this analysis of the properties of centrifugal governors, written by A. H. Church of New York University, is due Product Engineering, August, 1941.

Designed to maintain automatically constant speeds of prime movers, regardless of load demands, centrifugal governors are quite simple in operation. A centrifugal force which acts on rotating weights is balanced by known forces, usually gravity for low speeds and springs for high speeds. Any change in the speed of the unit being governed will vary this centrifugal force. For equilibrium to be maintained, the weights must assume a new position in response to any change of speed. This movement is used to vary the energy input to the machine being governed thus restoring its speed to the normal value. From this it is seen that a governor has two functions; to measure the prime mover speed and to vary the energy supply.

The properties of a governor are naturally dependent upon its design. Hence, before taking up the actual design procedure, it will be helpful to consider in some detail these properties and the way in which they can be measured. As outlined here, these properties agree with those given in the A. S. M. E. Power Test Code for Speed Responsive Governors.

Per cent Speed Regulation is the total speed variation of the machine expressed as a percentage of the mean or average speed of the unit This property may be measured for two conditions of operation:

Condition (A)—The change of speed which occurs with a load change.

Condition (B)—The departure from constant speed which occurs with constant load and constant operating conditions of the prime mover.

The regulation for which the governor is to be designed will depend upon the use of the driven machinery or apparatus. If the regulation becomes extremely small the governor is said to be isochronous. Under this condition the weights do not have a definite position for each speed, hence they will alternately move rapidly between the extreme positions, and the prime mover will experience sudden speed changes, known as "hunting."

In general, centrifugal governors have a regulation of 2 to 12 per cent; those operated in conjunction with an oil relay have a regulation of about half this amount. For Condition (B) the departure from a constant speed with constant load and operating conditions is small; the maximum value being about ¼ per cent. The regulation for this latter condition is more dependent upon the size of the rotating parts or a flywheel, than upon the governor; since the action is too rapid and frequent for governor control.

When N_a = RPM. of machine at high speed divided by 1000

N_b = RPM. of machine at low speed divided by 1000

the formula for determining the per cent of speed regulation R is

$$R = \frac{N_a - N_b}{(N_a + N_b)\,2} \times 100$$

$$= \frac{2\,(N_a - N_b)}{N_a + N_b} \times 100 \qquad (1)$$

The percentage regulation of a given unit can be determined by test. For Condition (A) the unit is operated at various loads, then after conditions have become steady, the exact speed is noted. A curve can then be plotted of speed against load, which will show any deviation from the desired straight line relationship. The regulation is usually based upon the two extremes of loading, that is, upon the no-load speed and full-load speed. For Condition (B) the extremes of speed are found by means of a sensitive

tachometer or oscillograph. Naturally, the regulation for this condition will be closer for a rotating machine than for a reciprocating one, since the energy input is more uniform.

Per cent Speed Adjustment is the percentage of the nominal or normal speed of the unit at which the governor will operate satisfactorily without exceeding acceptable values of regulation. For example, a turbine may be specified to operate at 2000 RPM. with a $\pm$ 10 per cent speed adjustment. This means that the governor should be adjustable so that the turbine will run satisfactorily at any speed between 1800 and 2200 RPM. This can be done by adjusting the tension in the outside spring.

Sensitiveness is a measure of the minimum change in speed at any governor position which will cause a movement of the governor mechanism. This property is measured as a percentage of the average or mean speed, hence can be calculated by Equation (1). Sensitiveness is a measure of the amount of friction and lost motion in the governor mechanism, and depends upon:

1. Design, that is, the number and type of pivots or working joints.
2. Lubrication of the moving parts.
3. Amount of vibration of the governor.
4. Forces transmitted to the governor through the sleeve.

Sensitiveness can be measured by slowly changing the load on the prime mover and observing the speed change required to cause the governor mechanism to change its position.

If there is too much lost motion or friction, no motion of the governor will occur until the speed change is large, and the mechanism will then move considerably after it does start. The result will be large swings in the speed, giving an undesirable hunting condition.

A governor with a small regulation will be more sensitive than one with a large regulation, as the movement of the mechanism will be greater for a given speed change.

Rapidity is a measure of the speed with which the mechanism of the governor adjusts the valves in response to a sudden load change. This property includes the speed variations which occur with sudden load changes and the time required for the prime mover to return to its normal speed.

In order to determine the rapidity it is necessary to use a recording tachometer or oscillograph which will record the speed of the prime mover from the instant the load is changed until the speed is again normal.

A governor with a small regulation will have a relatively small motion for a given speed change, so that the action of the governor in restoring the prime mover to the normal speed is rapid.

Stability. For each given speed of the prime mover there should be a definite position of the governor mechanism. If this is so, the governor is said to be stable. However, if the mechanism assumes a number of positions for a given speed, the governor is said to be unstable and tends to hunt. A stable governor generally has a low sensitivity and a large regulation, and is also rapid in action.

Governing Force or Strength is the regulating force of the governor, or the force available to operate the valves. Since it is a property of the governor, this force is always measured at the governor sleeve rather than at the valve.

If the speed of the prime mover is constant there is no force exerted on the valve levers, as the centrifugal force on the weights is balanced by the spring load or by the force of gravity. If, however, the governor sleeve is held fixed while the speed is changed 1 per cent, the force exerted on the sleeve is known as the strength, or ability to overcome unbalanced forces and friction. This property should be determined for both the inner and outer positions of the weights as the strengths will differ for the two positions.

The governor strength for a 1 per cent speed change can be determined by operating the prime mover at no load and 99.5 per cent of the normal speed. The linkage between the governor and valve is adjusted so that the weights will be in their high speed position. If the supply valve is then moved until the prime mover speed is 100.5 per cent of the normal and the governor sleeve held fixed, the resulting force exerted at the sleeve will be the strength. This test should be repeated with the weights in their low speed position.

The strength can be calculated for any given speed, N, as it is the difference in spring loads for a 1 per cent speed change. In order to derive this equation let

V = volume of weights, cu. in.

d = dia. through center of gravity of weights, in.

G = moment arm of center of gravity above the pivot point, in.

H = moment arm of spring load from pivot point, in.

Q = lever ratio = G/H.

N = speed in 1000 RPM., RPM./1000.

k = weight constant, lb. = $4.025\ VdQ$.

L = spring load, lb.

$$\text{Initial load on spring} = 4.025\ VdQ\ (0.995\ N)^2 = k\ (0.995\ N)^2$$

Final load on spring, assuming that the position of the mechanism is not changed

$$= k(1.005N)^2$$

$$\text{Available force} = k[(1.005N)^2 - (0.995N)^2] = 0.02kN^2 = 0.02L \qquad (2)$$

It is evident from Equation (2) that the strength for a given speed can be increased by increasing the size of the weights, the diameter to their center of gravity, or the lever ratio. Equation (2) does not include frictional effects in the governor itself; hence the equation will not check the test values exactly. The strength is limited by the size of the governor which in turn is limited by stress considerations and the size of the prime mover.

Work capacity for a 1 per cent speed change is the arithmetical mean of the governor forces at the inner and at the outer positions of the weights for a 1 per cent speed change multiplied by the arithmetical mean of the two corresponding sleeve travels for a 1 per cent speed change.

Any factor which will increase the strength will in turn increase the work capacity. The work capacity will also be increased by a longer travel for a given regulation, or smaller regulation for a given travel.

Design Procedure.

A sectional drawing of a steam turbine governor with its lever arms and outside spring is shown in Figure 1 to illustrate a typical centrifugal governor.

Centrifugal Force. If the density of steel is taken as 0.283 pounds per cubic inch, the centrifugal force F in pounds exerted by a particle of weight w pounds rotating about the center line at a radius r inches or a diameter d inches, with an angular velocity of ω radians per second, is

$$F = mr\omega^2 = \frac{wr}{g12}\left(\frac{2\ \pi\ 1{,}000\ N}{60}\right)^2$$

$$\text{or } F = 8.05\ V\ r\ N^2 = 4.025\ V\ d\ N^2 \qquad (3)$$

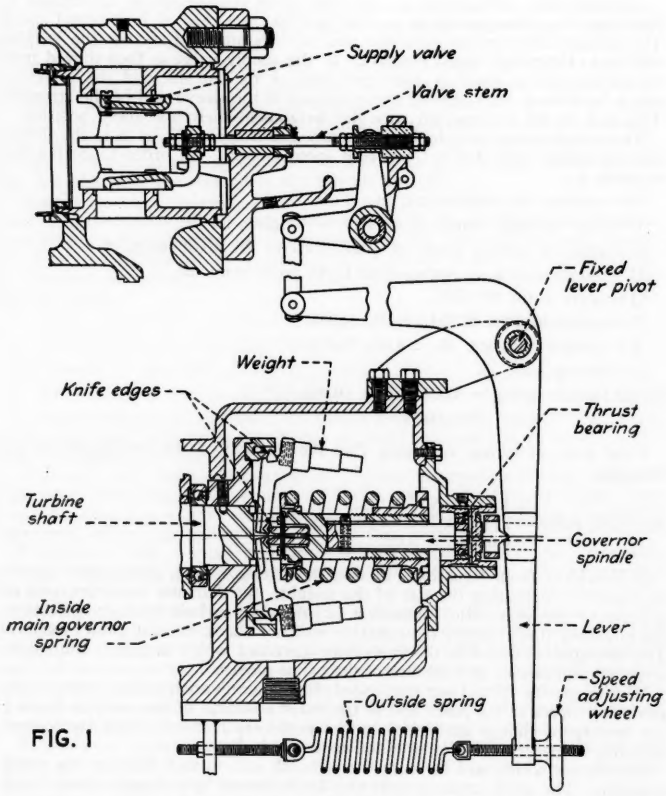


FIGURE 1.—TYPICAL CENTRIFUGAL GOVERNOR WITH LEVER ARMS AND OUTSIDE SPRING FOR GOVERNING A STEAM TURBINE. (DE LAVAL STEAM TURBINE COMPANY).

The schematic sketch, Figure 2, of a typical spring loaded centrifugal governor shows the forces acting. An analysis of these forces shows that for any position

$$L = \frac{FG}{H} = FQ = 4.025\ V\ d\ N^2 Q \qquad (4)$$

where V is the total volume of two weights. Since for any given position of a particular governor V, d and Q are always the same, Equation (4) may be written.

$$L = k\,N^2$$

FIG. 2

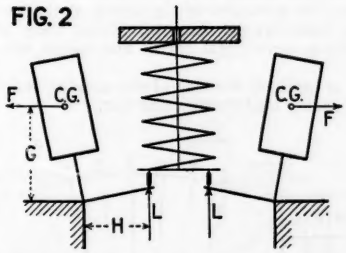


FIGURE 2.—SCHEMATIC DIAGRAM OF A TYPICAL SPRING LOADED CENTRIFUGAL GOVERNOR SHOWING THE FORCES IN THE SYSTEM.

The factor k is a function of the moment of the centrifugal force as applied at the spring seat.

In terms of the spring load for the final or outer weight position L_t in pounds the spring load for the initial or inner weight position L_o in pounds, and the sleeve travel T in inches; the scale S of the governor spring in pounds per inch is given by

$$S = \frac{L_t - L_o}{T} \qquad (5)$$

It can be shown (see pages 448-449 of the first edition of Mechanics of Machinery by Ham & Crane) that it is not correct to find the centrifugal force moment by assuming the mass of the weights and lever arms as concentrated at their centers of gravity. Instead, these items should be broken down into small simple shapes, and the moment found for each separately. These moments may then be summed up to obtain the total moment for any position of the mechanism. The correct method of obtaining this moment would be to employ integration, but it is usually impractical. The need for breaking up the masses into small parts increases when the line through the pivot and center of gravity of the whole mass makes a large angle with the center line of rotation, and when the distance between this center of gravity and the pivot point is large, that is, for "long" weights.

Governor. To illustrate the design procedure simply it will be assumed although incorrectly as just noted, that the mass of the weights is concentrated at their centers of gravity as shown in Figure 2, and that the weight of the arms is neglected. The nominal speed N, the desired percentage regulation R and the travel of the sleeve T are given. The volume of the two weights V is found by trial to give the desired strength, and the dimensions of the mechanism assumed with regard to the overall size of the governor. From these preliminary values a layout, which should be several times full size for accuracy, and a table as shown in Figure 3 are made.

The total travel T is divided into a number of equal steps, and the corresponding positions of the center of gravity located as shown. The values of H, G and d are listed in the table for the various positions. From these the lever ratios Q and the factors k can be obtained.

Knowing the nominal speed N and the per cent regulation R, the values of N_o and N_t can then be determined and entered in the table. The values of centrifugal force F, and spring load L, can then be calculated using the spring loads at the end positions. The required spring scale can be calculated by the Equation (5). It should be noted that these spring loads and this spring scale are really "equivalent" as will be explained later.

Having determined the spring scale S, which is constant, the end spring loads L_t and L_o, and the deflection of the spring at the various positions, the intermediate spring loads can be obtained. From these intermediate spring loads the corresponding centrifugal forces and speeds are found, thus completing the table.

In the procedure as outlined, the size of the weights and lever ratio may be varied until desired spring load condition is met.

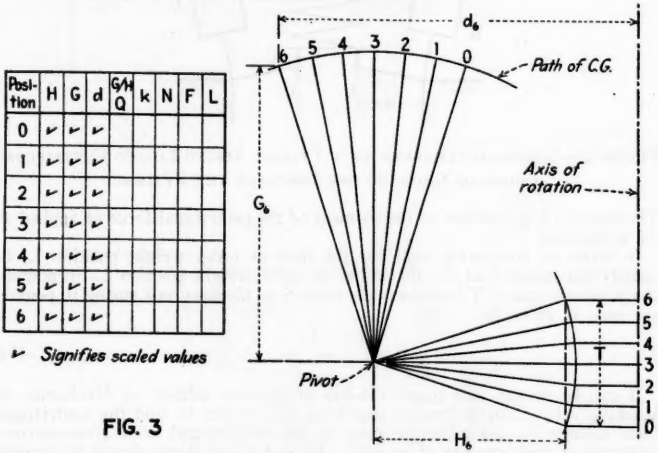


FIGURE 3.—GRAPHICAL CONSTRUCTION FOR OBTAINING VALUES OF MOMENT ARM OF SPRING LOAD FROM PIVOT POINT, MOMENT ARM OF CENTER OF GRAVITY ABOVE THE PIVOT POINT, AND THE DIAMETER OF THE PATH OF THE CENTER OF GRAVITY.

Operating Leverage. A sketch of the leverage connecting the governor with the energy supply to the prime mover is shown diagrammatically in Figure 4. As mentioned previously, the values of spring load and spring scale found in the procedure described are "equivalent" since they represent the outside spring load L_1 and the outside spring scale S_1, combined with load L_2 and the scale S_2 of the main governor spring.

Where T is the travel of the governor stem, it is seen from Figure 4 that the travel of the supply valve is given by $T \times (c/a)$, and the deflection of the outside spring by $T \times (b/a)$. All forces and scales are referred back to the governor, so that expressions can be derived for the equivalent spring load E_L in pounds, and the equivalent spring scale E_s in pounds per inch, with both terms referred to the governor sleeve, so that

$$E_L = L_2 + L_1 \times (b/a) \tag{6}$$

$$E_s = S_2 + S_1 \times (b/a)^2 \tag{7}$$

The equivalent spring scale E_s and the equivalent spring load E_L must be divided between the outside and inside springs in accordance with Equations (6) and (7). It is unwise to put too much load on the governor thrust bearing by making the outside spring load or scale too large.

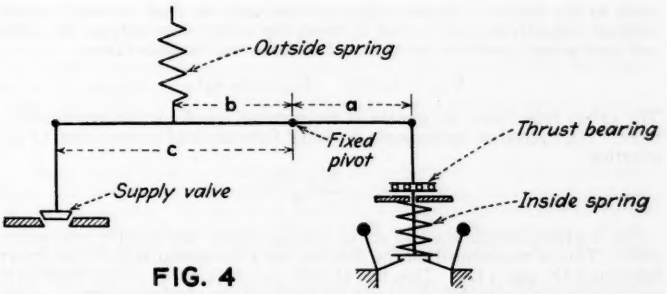


FIGURE 4.—DIAGRAMMATIC SKETCH OF THE LEVERAGE SYSTEM CONNECTING THE CENTRIFUGAL GOVERNOR WITH THE ENERGY SUPPLY VALVE FOR THE PRIME MOVER.

Speed Adjustment. It may be desirable to have a certain range of speed through which the unit will act satisfactorily, that is, speed adjustment. This may be obtained, within limits, by altering the initial load on the outside spring.

For example, suppose that the unit is to operate with a nominal speed 10 per cent greater or less than its usual value N without having its regulation exceed the limits of 2 to 10 per cent. For a given mechanism and weights, curves like those shown in Figure 5 can be plotted. With low speed N_b and high speed N_a being the nominal speeds for the given adjustment plus or minus one-half the regulation R, E_{Lo} and E_{Lt} being the equivalent spring

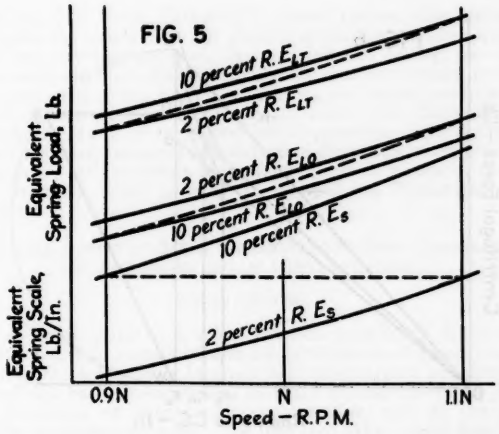


FIGURE 5.—GRAPHS FOR DETERMINING FACTORS NEEDED TO CALCULATE SPRING LOAD AND SCALE VALUES.

loads at the initial or inner weight position and the final or outer weight position respectively, also k_o and k_t being the weight constants at the initial and final weight positions, the spring load curves can be found from

$$E_{L.o} = k_o N_b^2, \quad E_{L.t} = k_t N_a^2$$

The values thus found are plotted at the nominal speed for the given adjustment. The equivalent spring scale is plotted from the load points found by the equation

$$E_s = \frac{E_{L.t} - E_{L.o}}{T}$$

For a given mechanism and set of springs, there can be only one spring scale. This is represented on the diagram by a horizontal dotted line drawn between 0.9N and 1.1N. This line should not cross either of the equivalent spring scale curves as they represent the limits of the regulation range. After this equivalent spring scale has been established, the corresponding points on the E_{Lo} and E_{Lt} curves are found. These curves are also shown dotted on the figure. There is no reason for attempting to find any points other than the end points, as they are all that are needed for designing.

Having obtained E_s, E_{Lo} for 2 and 10 per cent regulation and E_{Lt}, for 2 per cent regulation from the curves, the spring scales and loads for the inside and outside springs can be found by the Equations (6) and (7). It is desirable to have the initial outside spring load zero or nearly so for 0.9N and the 10 per cent regulation, and to keep the outside spring scale as low as possible so as to reduce the load on the governor thrust bearing. Having found the loads and scales of both springs, they can now be designed to suit space and stress considerations.

Characteristic Curves. If the centrifugal force acting on the weights is plotted against the radial position of their center of gravity as shown in Figure 6, more information concerning the governor action is disclosed. Referring to this figure, the curve bma is the theoretical curve for a typical governor.

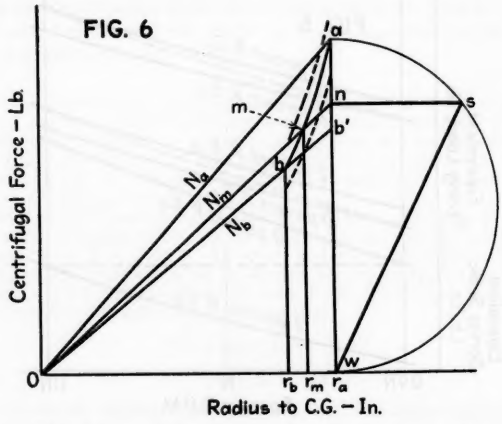


FIGURE 6.—CHARACTERISTIC CURVE FOR DETERMINING THE PROPERTIES AND ACTION OF A CENTRIFUGAL GOVERNOR.

There will be a certain amount of friction and lost motion in the mechanism so that the actual curves will be displaced somewhat from the theoretical. These are shown as dotted lines in the figure, the upper line representing the actual curve obtained when the weights are moving outward and the lower line for the weights moving inward.

The radial lines oa, on and ob′ are lines of various constant speeds since

$$F = 8.05\ V\ r\ N^2 = \text{constant} \times r$$

If these radial lines coincide with the characteristic curve for short distances, the weights may assume several positions for a given speed and the governor is unstable. If the curve should be a straight line which if extended would pass through the origin O the governor would be isochronous. Hence, the larger the angle which the curve makes with the radial lines, the more stable the governor.

It is shown in Trinks "Governors, and the Governing of Prime Movers," that the percentage regulation is given by the value of the expression

$$\frac{ab'}{aw + b'w}$$

By drawing the semi-circle asw and ray omn to aw and the horizontal ns

$$N_m = \frac{ws}{wa} N_a$$

so that if the speed at a point such as a is known, the speed at any other point on the curve such as m can be found graphically.

PHYSICS OF STEAM GENERATION AT HIGH PRESSURES.

This paper by R. F. Davis, Technical Engineer, Internal Combustion, Limited, investigates the constitution and physical properties of boiler water with reference to the operating pressure. Nuclear and film boiling, size of steam bubbles generated and their effect on circulation, relation of foaming to physical state of suspended solids, mechanical priming and steam contamination by moisture and solids are among the subjects discussed. Mr. Davis' paper, here reprinted from the March, 1941, Journal and Proceedings of the Institution of Mechanical Engineers, occasioned considerable discussion when presented at London and Manchester meetings of the Institution. A portion of the discussion, taken from the April issue of the Journal and Proceedings, is also reprinted below.

The Physical Aspect of Steam Generation at High Pressures and the Problem of Steam Contamination.

Boiler Water.

In general, boiler water is a dilute aqueous solution of dissolved salts, together with matter in suspension and traces of dissolved gases. As evaporation proceeds, the boiler water becomes more concentrated, and precipitation occurs. Each salt has its own peculiar solubility characteristics, its precipitation point depending on temperature, concentration, and the quantity and nature of the accompanying salts. The dissolved salts consist mainly of the sodium salts: sodium sulphate, sodium chloride, and sodium hydroxide. The first, sodium sulphate, may occur up to 75 per cent, and the other two salts in amounts up to 50 per cent of the total dissolved solids.

In high-pressure and central station plants, the amount of dissolved solids is usually kept well below 3000 p.p.m.,* but in low-pressure industrial plants the concentration may be allowed to rise to nearly 10,000 p.p.m. It has been recommended (Glinn 1935) that 7000 p.p.m. is the limit of good practice up to 600 pounds per square inch, and 1500 p.p.m. above 850 pounds per square inch up to 1500 pounds per square inch.

The effect of the dissolved salts on the surface tension depends on the degree of adsorption, and is given by Dorsey's formula (Kaye and Laby 1935)

$$\gamma_c = \gamma + KC \qquad (1)$$

where γ_c, γ, are the surface tensions of the solution and water respectively, at the same temperature, K is a constant, and C is the concentration of the bulk of the solution. Although the value of K may vary within the range of 1.53 and 2.00, according to the nature of the salt, it will be sufficiently accurate to take a mean value K = 1.77. Again, for the three commonest salts in boiler water, taken at a concentration of 1000 p.p.m. (i.e., 1 gm. per liter), the concentration C in gram equivalents per liter is

NaCl	$C = 1/58.5$
$\frac{1}{2}(Na_2SO_4)$	$C = 1/71$
NaOH	$C = 1/40$

Or, for the average boiler water, say, $C = 1/56.5$, and $KC = 1.77/56.5 = 1/32$.

The surface tension of boiler water having normal concentrations is accordingly given by the approximate formula

$$\gamma_c = \gamma + c/32{,}000 \qquad (2)$$

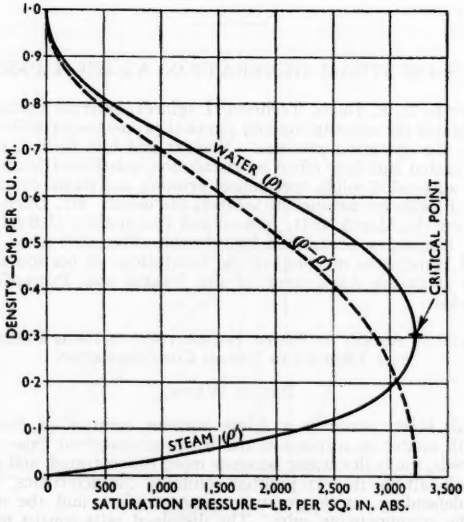


FIGURE 1.—DENSITY OF SATURATED STEAM AND WATER.

* P.p.m. parts per million.

where c is the total concentration of dissolved solids in parts per million. Since the surface tension γ of pure water in contact with its vapor is 71.4 dynes per centimeter at 15 degrees C. (Kaye and Laby 1935), it follows that

$$\gamma_c = 71.4 + 10/32 = 71.7 \text{ dynes per cm.}$$

for the extremely high concentration value of c = 10,000 p.p.m. This is of interest as demonstrating that the surface tension is not greatly affected by the concentrations met with in practice, for $\gamma_c = 71.7$ dynes per centimeter exceeds the corresponding figure for pure water by less than 0.5 per cent. There is, however, considerable variation in the surface tension, depending on the saturation densities of the water and the steam, as may readily be proved by means of Macleod's formula (see Appendix I, equation (15)). In view of the fact that the surface tension of water in contact with its vapor is 71.4 dynes per centimeter at 15 degrees C., it is thus seen that for higher saturation temperatures.

$$\gamma = 71.4(\rho - \rho')^4 \tag{3}$$

where ρ and ρ' represent respectively the densities of the water and its vapors, in grams per cubic centimeter, at a given saturation temperature (or pressure). The densities are shown in Figure 1, and the corresponding surface tensions in Figure 2.

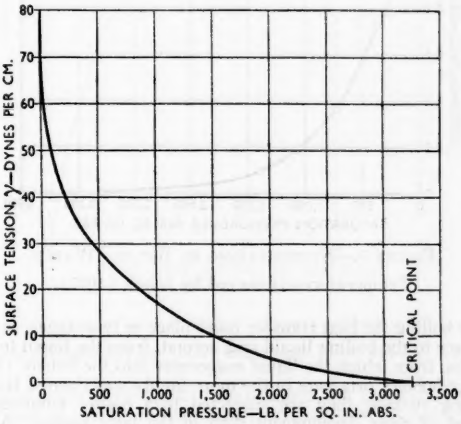


FIGURE 2.—SURFACE TENSION OF WATER IN CONTACT WITH STEAM.

$$\gamma = 71.4(\rho - \rho')^4$$

STEAM GENERATION.

Evaporation at a Heated Surface.—Evaporation from a submerged heated surface may take place either by nuclear boiling, or by film boiling. In nuclear boiling the water evaporates into steam bubble spaces and the water is, therefore, superheated because the surface tension at the bubble interface raises the internal pressure. Theoretically, nuclear boiling cannot originate at an absolutely plane surface, where the initial diameter of a bubble would be

infinitely small and the superheating of the water infinitely high, but in a boiler the metallic surface in contact with the water has free binding forces, which may build up small solid particles and gases into layers of from 10 to 20 molecules in thickness. A surface indentation or scratch will accordingly be associated with a highly concentrated mass of adherent particles, and form a nucleus for evaporation if the metal is heated above the boiling point of the liquid (Schmaltz 1937). Surface irregularities and roughness due to oxide and scale deposits can also become points of nuclear generation for the same reason.

In nuclear boiling it is the slight superheating of the water which provides the impulse for heat transfer from the water to the steam. The amount of this superheating varies with the pressure, and the values found by Jakob (1937) and his co-workers for water in bulk are shown plotted in Figure 3. Moreover, the amount of superheating decreases with increasing pressure because the surface tension decreases; in fact Figure 3 closely resembles Figure 2. In close proximity to the heating surface, the temperature of the water approaches the temperature of the surface itself.

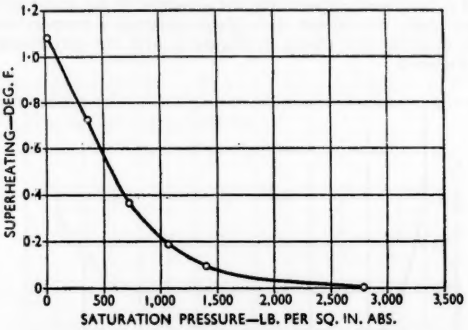


FIGURE 3.—SUPERHEATING OF BOILING WATER.
Temperatures Observed by Jakob (1937).

In nuclear boiling the heat transfer takes place in two stages, first, from the heating surface to the boiling liquid and, second, from the liquid to the surface of the bubbles, from which the liquid evaporates into the bubble (Bosnjakovic 1930). Since bubbles originate in the more highly superheated layer adjacent to the heating surface, they are subjected to a higher temperature in the earlier period of their development than in the later stages. According to Jakob the coefficient of heat transfer across the bubble surface at the moment of origin of the bubble is of the order of 40,000 Btu. per square feet per hour per degree F. temperature difference, but this is only momentary, and decreases rapidly as the bubble grows in size. When the bubble has detached itself from the heating surface and so left the highly superheated layer, the transmission rate across its surface tends to assume a steady value of about 3000 Btu. per square foot per hour per degree F. The temperature difference is that between the superheated water and the steam in the bubble.

The temperature difference between the heating surface and the water as a whole is affected to a certain extent by the convection or stirring effect of the rising bubbles and, therefore, by the number of nuclei per unit area of heating

surface and by the frequency with which bubbles form at a given nucleus. Jakob found the coefficient of heat transfer between a heated metal plate and water to vary, not only with the intensity of heating, but also with the condition of the surface, that is, its roughness and whether it had been previously exposed to the atmosphere and had adsorbed air on its surface, also with the length of time it had been in service. Figure 4 is reproduced from Jakob's results for a horizontal metal surface sand blasted and then oxidized by long use, and probably represents the condition of a boiler plate or tube in service more exactly than other graphs given by the same investigator. Graphs of much the same shape were obtained with a specified heating surface when placed vertically or horizontally.

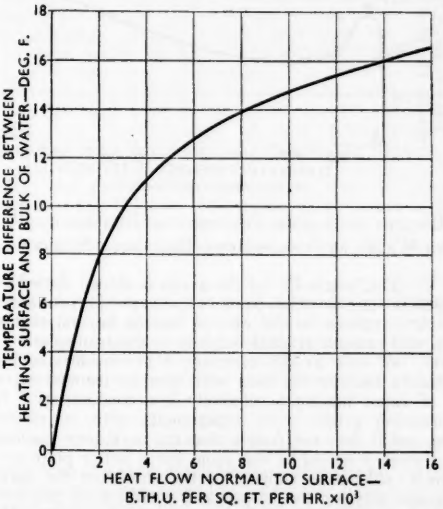


FIGURE 4.—TEMPERATURE OF METAL HEATING SURFACE (JAKOB).
Surface Roughened by Sand Blast and Oxidized by Use.

Once a state of nuclear boiling has been attained at a given place, the steady application of heat at that place results in a uniform rate of bubble formation there, but an increase in the rate of heating gives rise to fresh nuclei because the higher superheating of the water in contact with the heating surface allows vapor bubbles to originate at places of smaller curvature. This process does not, however, continue indefinitely, for, with an increasing rate of heating, a state is eventually attained in which the bubbles become so numerous and closely packed that they coalesce and so form a continuous film of vapor between the heating surface and the liquid. Then evaporation takes place by film boiling, when the vapor leaves the film in the form of large irregular shaped bubbles.

The change from nuclear to film boiling is well illustrated by the photographs obtained by Drew and Mueller (1937) in their experiments on the boiling of carbon tetrachloride. In the same paper reference is made to the curve for

water derived by Nukiyama (1934), which is reproduced in Figure 5, in order to show the quantitative effect of a progressive increase in the rate of heating. It is clear that the difference of temperature between the heating surface and the water is not very great between the limits defined by nuclear boiling (A to B on the graph). At the point B, however, this law of variation undergoes a sharp change, of a kind such that further increase in the heat flow will actually lead to a decrease in the transmission rate, until the minimum value

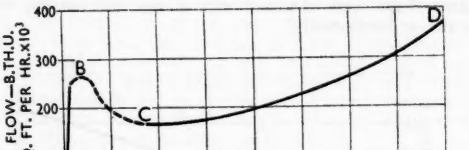


FIGURE 5.—BOILING RATE AND TEMPERATURE DIFFERENCE BETWEEN METAL AND WATER AT ATMOSPHERIC PRESSURE (NUKIYAMA).

is reached at C. The length BC of the graph is shown dotted because it represents an unstable state in which parts of the vapor film collapse in an irregular manner. Any increase in the rate of heating beyond that corresponding to the point C will cause a gradual increase in the transmission rate (C to D on the graph), but only at the expense of a considerable increase in the metal temperature because the heat will then be transferred across a continuous film of vapor having a relatively low conductivity. Nukiyama derived the foregoing graph from experiments with an electrically heated platinum wire, and it does not follow that the maximum nuclear boiling rate indicated by Figure 5 would be the same for a boiler plate or tube. There is, indeed, much evidence in support of the view that the maximum rate is influenced considerably by the nature and condition of the heating surface. For instance, Jakob obtained heat flow rates as high as 95,500 Btu. per square foot per hour from a flat horizontal copper plate, and at this rating nuclear boiling still persisted. Again, with a vertical cylindrical heating surface of the same material, he found that above 74,000 Btu. per square foot per hour the transmisison rate from the heating surface to the water decreased with a rapid rise in the metal temperature, which eventually burnt out the electric heating elements. The latter rating for a vertical surface corresponds to the heat flow which results in a boiler tube when subjected to the radiation from a luminous flame at a temperature of about 2300 degrees F.

That the condition of the heating surface does have considerable influence on the critical rating, at which the change from nuclear to film boiling occurs, has also been demonstrated by the experiments of Rhodes and Bridges (1938), who found that the wettability of the surface, or more accurately the contact angle between the liquid and the heating surface, has a profound effect on the critical rating. Relatively small amounts of certain substances in solution or in suspension may also have a great effect in altering the manner of boiling. Evidently the critical rating for a boiler tube or plate will depend on the nature of the boiler water, on the condition of the effective heating surface, and, possibly, on the inclination of the surface.

Normally evaporation in a water tube boiler takes place by nuclear boiling, since film boiling would obviously involve such high metal temperatures that the tubes would quickly fail by rapid creep or plastic flow of the metal, especially with the high saturation temperatures which accompany high pressures.

In most cases where tube failures have been reported as due to flame impingement, it is possible that either scale or choking of the tubes by sludge deposit has been a contributory cause, but the possibility of overheating of the metal under conditions of film boiling cannot be ruled out entirely until more is known about the subject.

Investigations recently carried out in America by Partridge and Hall (1939) have disclosed a phenomenon analogous to film boiling. This "steam blanketing" has, in the main, been found to occur in the case of boilers operating above 1200 pounds per square inch and having tubes that are insufficiently inclined to the horizontal. At high rates of heat input, the bubbles collecting at the crown of these nearly flat tubes form an almost continuous insulating steam blanket. The consequent overheating of the metal at the crown of the tube and its intermittent splashing by the relatively colder water resulted in some instances in cracking of the metal, and in others the localized concentration of the boiler water caused erosion. A state of bubble-stagnation at the crown of slightly inclined tubes is usually prevented by the circulation of the water. In the majority of the cases mentioned by Partridge and Hall failure occurred in very flat ash or slag screen tubes and the tubes supporting slagging furnace bottoms. One of the conclusions to be drawn from their investigations is that the tubes of high-pressure boilers exposed to high gas temperatures should have an inclination of at least 12½ degrees with the horizontal.

The Formation of Steam Bubbles at a Heated Surface.—The initial size of a steam bubble depends on the radius of curvature of its original nucleus, which may be a minute gas bubble, or a particle of sludge or scale adhering to the surface, or a mere roughness on the heating surface. If the initial nucleus is a solid spherical surface, it cannot become available for growth until the vapor pressure at that point exceeds the equilibrium state, giving

$$r = \frac{2\gamma}{p_1 - p_0} \tag{4}$$

as the minimum nuclear radius available with an external pressure p_0 and an internal vapor pressure p_1 in a liquid having surface tension γ. Surfaces with much larger radii of curvature than that given by equation (4) do not become nuclei, because they do not provide the same degree of concentration of vapor molecules per unit of projected area within the same period of time.

If the initial nucleus is a small gas bubble, the conditions will be modified by the partial pressure within the bubble. It can be shown (Newman and Searle 1933) that a gas bubble becomes available as a nucleus when

$$r = \frac{4}{3}\frac{\gamma}{p_1 - p_0} \tag{5}$$

Table 1, calculated by means of equations (4) and (5), shows the minimum radii of gas bubbles and solid surfaces which become available at different degrees of water superheating under atmospheric conditions.

It is evident from Figures 4 and 5 that with nuclear boiling the heating surface is not likely to exceed the temperature of the bulk of the water by more than about 40 degrees F., under atmospheric conditions, and the superheating of the water layer in contact with the heating surface must be less than that figure. It will be noted that at 40 degrees F. superheat, the dimensions of the available nuclei approach those of particles which form perma-

TABLE 1. MINIMUM RADII OF GAS BUBBLES AND SOLID SURFACES

Superheat of water, degrees F.	Radius of gas bubble, microns	Radius of solid surface, microns
2	23	35
4	11	17
10	4.3	6.5
20	2.0	3.0
30	1.2	1.8
40	0.8	1.2

nent suspensions. This gives some idea of the extreme smallness of the nuclei. At higher pressures, due to the decrease in surface tension (see Figure 2) still smaller nuclei become available. Nuclear boiling, therefore, continues to exist at high pressures and with reduction in the degree of superheat (Figure 3).

The subsequent shape of the bubble, as it grows in size by evaporation from its surface, is determined by the forces acting upon it. When the heating surface is horizontal and faces upwards, the bubble is symmetrical about its vertical axis, and its shape and size are calculable from considerations of static equilibrium. To illustrate the point, let Figure 6 represent the axial

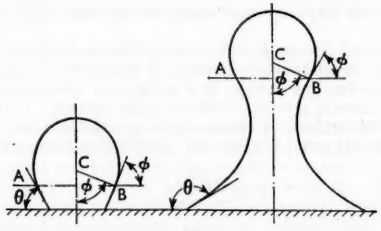


FIGURE 6.—DIAGRAM OF FORMATION OF STEAM BUBBLE.

section of a bubble anchored at the heating surface, AB any horizontal plane drawn through the bubble so that the tangent at A makes an angle ϕ with the horizontal, and let v be the volume of the bubble above AB. If r_1 is the radius of curvature in the plane of the diagram at B, and BC is the normal at B meeting the axis at C, then $BC = r_2$ is the other principal radius of curvature at B. The forces acting at the plane AB under static equilibrium, assuming the external pressure p_0 is sensibly uniform over the height of the bubble, are:—

$$\text{Upwards: } p_1 \pi r_2^2 \sin^2\phi + (\rho - \rho')vg$$

$$\text{Downwards: } p_0 \pi r_2^2 \sin^2\phi + \gamma \sin\phi 2\pi r_2 \sin\phi$$

which when equated give

$$2\gamma\pi r_2 \sin^2\phi = (p_1 - p_0)\, \pi r_2^2 \sin^2\phi + (\rho - \rho')vg$$

The terms p_0 and p_1 can be eliminated by combining the above expression with equation (16) in Appendix I, and substituting the contact angle

θ for ϕ in the resulting expression for v, to obtain the total volume V of the bubble above the heating surface, in the form

$$V = \frac{\pi\gamma \sin^2\theta}{(\rho - \rho')g}\left(\frac{1}{r_2} - \frac{1}{r_1}\right) r_2^2 \qquad (6)$$

Although the angle of contact has no influence on the shape of the bubble above the surface, it nevertheless determines the maximum volume above which the bubble becomes unstable and breaks away, under conditions in which the buoyancy exceeds the surface tension at the circle of contact with the heating surface. To illustrate this point the profiles of air bubbles in water are reproduced in Figure 7 for different initial radii of curvature.

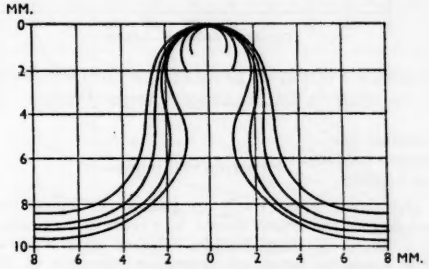


FIGURE 7.—SHAPES AND SIZES OF STATIONARY AIR BUBBLES IN WATER (WARK 1933).

These are mostly shown for the limiting value of $\theta = 180$ degrees; for lesser values of the contact angle a separate horizontal line must be drawn through each profile, so that each profile meets each line at the correct contact angle.

To determine the shape of a bubble and its maximum volume requires special methods of analysis, which Jakob and his co-workers, following the method of Wark, have reduced to a simple curve (reproduced in Figure 8) connecting the contact angle θ with the dimensionless expression $(V/a^3)^{\frac{1}{3}}$ where V is the maximum volume and

$$a = \sqrt{\frac{2\gamma}{(\rho - \rho')g}} \qquad (7)$$

This curve has been checked by Jakob for water over the range of contact angles θ from 20 degrees to 100 degrees by measuring the photographs of bubbles in the process of formation (Figure 18, Plate 2). Considering that the real problem is dynamical and not statical, as assumed in the theoretical analysis, the agreement is remarkably close.

Combining equation (7) with equation (3) and using the straight-line relationship of Figure 8 between θ and (V/a^3), a very simple formula can be established for the mean diameter of the steam bubbles, leaving a horizontal heating surface, namely

$$d = \frac{\theta}{17.8}(\rho - \rho^1)^{1.5} \qquad (8)$$

where d is the mean diameter (with respect to volume) of the bubbles in millimeters, θ is the contact angle in degrees, and ρ and ρ' are the densities

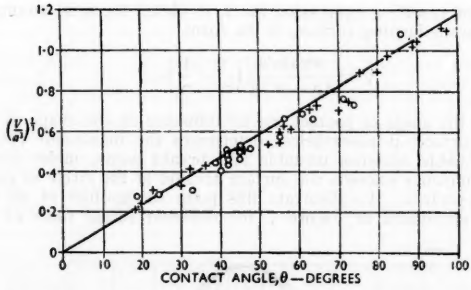


FIGURE 8.—MAXIMUM VOLUME OF BUBBLES AND CONTACT ANGLE, ACCORDING TO FRITZ'S CALCULATION (AFTER JAKOB).

——— Theoretical line.
\+ Hydrogen bubbles } Experimental points.
O Steam bubbles }

of the liquid and vapor respectively in grams per cubic centimeter. By means of equation (8) the chart shown in Figure 9 has been prepared, to show the mean diameter of steam bubbles released from a horizontal heating surface over the complete range of saturation pressure of steam, and for contact angles from zero to 90 degrees.

The contact angle only equals or exceeds 90 degrees with a very smooth or a greasy surface. In the case of contact angles greater than 90 degrees, the bubbles are liable to take up a re-entrant shape, as in the right-hand portion of Figure 6, and break away at the narrowest section of the neck, where it is nearly cylindrical, and the volume of the released portion is the same as if the contact angle were 90 degrees. Bubbles of a re-entrant shape have a considerable area of contact with the heating surface, thereby causing a rise in temperature of the surface. They also tend to join up with neighboring bubbles and form a continuous film more easily than when the contact angle is less than 90 degrees. Hence, at a greasy surface the change from nuclear to film boiling occurs at a much lower rate of heat flow, as found by Rhodes and Bridges.

Approximate measurements of the contact angle made by the author, between water and the surface of a piece of normal mild steel boiler tube, taken from a boiler which had been in service, gave a contact angle $\theta'' = 62$ degrees when the water was approaching the dry surface, and $\theta' = 20$ degrees when receding from the wetted surface. Probably the maximum bubble size is governed by the true contact angle θ, where $\cos\theta = \frac{1}{2}(\cos\theta'' + \cos\theta')$, whence $\theta = 45$ degrees. It is possible, however, that the contact angle measured under atmospheric conditions may not correspond to that existing during evaporation. The only satisfactory method would be to photograph steam bubbles in the process of formation; so far this has not been recorded for boiler water in contact with steel that has been in service, but provisionally it may be assumed that in a boiler the contact angle is about 45 degrees.

It has been demonstrated that in the case of a horizontal heating surface facing upwards the size of the steam bubbles at release is calculable, within limits, and varies with the boiler pressure and, to some extent, with the nature of the boiler water and the condition of the heating surface. In practice, heating surfaces have non-uniform inclinations, and the size of bubbles at

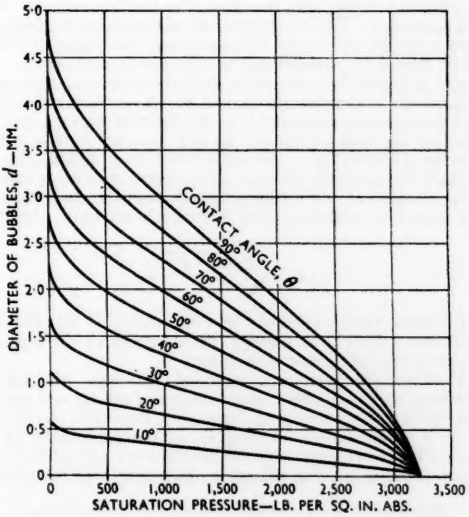


FIGURE 9.—MEAN DIAMETER OF STEAM BUBBLES LEAVING A HORIZONTAL HEATING SURFACE.

$$d = \frac{\theta}{17.8}(\rho - \rho^1)^{1.5}$$

release, though observable, is not amenable to mathematical treatment. For a heating surface facing upwards, however, the size of the bubbles released does not change to any appreciable extent between the limits of the horizontal and vertical positions of the surface, but when the surface is facing downwards the results are different; the more nearly horizontal the surface is, the larger the bubbles become before being released, and they tend to flatten out against it, causing, in the extreme case, steam blanketing and possible overheating of the metal.

In the tubes of a water-tube boiler the bubbles are subjected, in addition to their buoyancy, to a transverse shearing force imparted by the flow of the water along the tube. Bubbles are thus torn off the heating surface more quickly and are therefore smaller in size than if detached by the action of buoyancy alone. In a water-tube boiler, therefore, at release from the heating surface some bubbles will be larger and some will be smaller than indicated by Figure 9, depending on the velocity of circulation in each tube, and the slope and aspect of each individual portion of the heating surface.

The Migration of Steam Bubbles from the Heating Surface to the Free Water Level.—At the instant of release of each bubble from its circle of contact with the heating surface (Figure 18), the narrow cylindrical neck which forms between the base of the bubble and the heating surface, rapidly elongates and closes in, with an impact, causing a sound wave. The characteristic noise associated with ebullition is produced by a succession of these minute impacts, which occur with a momentum that largely depends on the viscosity of the liquid. At higher saturation pressures the viscosity of water

diminishes (Sigwart 1936) and, therefore, the suddenness with which bubbles are released increases. The frequency of release from a given nucleus will also increase with the saturation pressure and, according to Jakob's experiments, will be inversely proportional to the bubble diameter.

Immediately a bubble is released, it will commence to rise on account of its buoyancy, at an increasing speed, and the downward pull of the surface tension, which previously anchored it to the heating surface, will be replaced by a pressure on the upper surface. In this case the bubble will flatten out to a lens shape (Figure 18). In sufficiently deep liquid, the bubble will gradually attain its terminal velocity and approximate to a spherical shape. Generally the conditions in a boiler are such that the motion of the bubbles through the liquid is turbulent, and Rittinger's formula (Martin 1928) is applicable

$$U = 12.35 \sqrt{\frac{(\rho - \rho')}{\rho}} d \qquad (9)$$

where d is the mean bubble diameter in millimeters, U the terminal velocity in centimeters per second, and ρ and ρ' the densities of the liquid and vapor respectively, in grams per cubic centimeter.

Figure 10 shows the calculated terminal velocities for bubbles leaving horizontal heating surfaces with contact angles of 30, 60, and 90 degrees, and having mean diameters according to Figure 9. Equation (9) is based on the assumption of a single bubble moving in an extensive field of liquid, and Jakob records a mean velocity of 26 centimeters per second for bubbles from 1 millimeter to 8 millimeters in diameter; this agrees fairly well with equa-

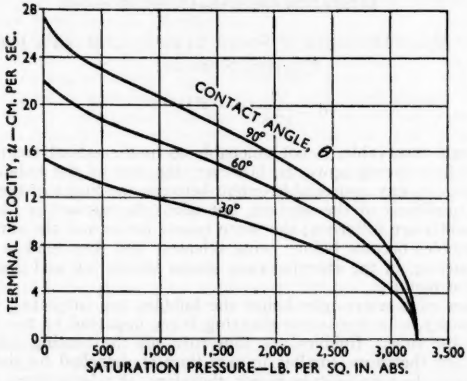


FIGURE 10.—TERMINAL VELOCITY OF STEAM BUBBLES LEAVING A HORIZONTAL HEATING SURFACE.

$$U = 12.35 \sqrt{\frac{(\rho - \rho')}{\rho}} d$$

tion (9), which for the same range of sizes gives a mean velocity of 24 centimeters per second. On the other hand, Behringer (1934) found that when the bubbles are closely packed in a tube, they rise at considerably greater velocities, as indicated by the curves in Figure 11, which are reproduced from his experimental results.

Behringer's results merit close consideration, because in a water-tube boiler the size and relative velocity of the bubbles control the circulation and, to some extent, the quality of the steam. Behringer attributes the increase in the relative velocity of the steam and water with increase in the proportion of steam, to the mass-effect of the closely packed bubbles and the entrained liquid. Yet an exactly opposite phenomenon occurs with a cloud of solid particles moving through a liquid; it is well known that the rate of sedimentation of solid particles decreases with increase in the concentration. It is strange, therefore, that bubbles seemingly behave in a contrary manner; the more concentrated they are, the faster they move, and it is just possible that Behringer's explanation is not correct.

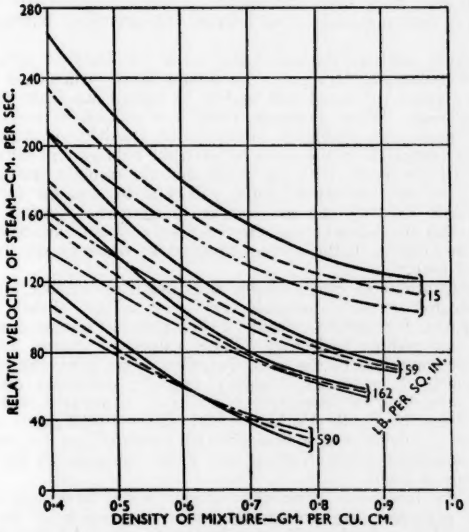


FIGURE 11.—VELOCITY OF STEAM BUBBLES RELATIVE TO WATER IN A VERTICAL TUBE (AFTER BEHRINGER).

——— Tube 3¼ inches outside diameter.
- - - - Tube 2¾ inches outside diameter.
——.—— Tube 2¼ inches outside diameter.

The tube which Behringer used was totally enclosed and presumably he had no means by which he could see what was happening inside it. Brooks and Badger (1937), experimenting with a long glass tube heated throughout its length, observed that the bubbles coalesced, pushing ahead of them slugs of liquid, which subsequently became thinner by evaporation and finally broke, allowing the vapor to pass through them. Annular rings of liquid then appeared, and spread over the walls, in the form of an irregular film.

Although the tube used by Behringer was not heated throughout its length, as in Brooks and Badger's experiments, it is nevertheless probable that some

of the bubbles, in passing up the tube, coalesced with others. Presumably Behringer, and also Brooks and Badger, used in their experiments pure water, or water containing only a very small proportion of dissolved salts. The bubbles in pure or nearly pure water will easily coalesce, and because of this, pure water will neither aerate nor foam. It is extremely likely, then, that coalescence did occur in Behringer's steel tube, as was observed in Brooks and Badger's glass tube.

In a boiler, coalescence of the steam bubbles is not so likely to occur. It is known, for instance, that under certain conditions boiler water will form a stable foam, and this property can usually be attributed to the influence of either suspended solids or colloidal matter; but a liquid can aerate, or possess the property of "foamability," without necessarily forming a stable foam. Dissolved salts alone can impart to water the property of aeration, or foaming in such a manner as to prevent coalescence of bubbles below the water level.

Edser (1922) explains the stabilizing action of dissolved salts on bubbles in terms of molecular attraction. In a pure liquid, when two bubbles approach one another the liquid will tend to be sucked out from between them and they coalesce. When, however, there is a definite increase or decrease in surface tension with change in concentration (positive or negative adsorption), the concentration in the space between the bubbles will differ from that of the bulk of the liquid, with the result that the osmotic pressure becomes unbalanced, and the unbalanced force will tend to stop the motion of the bubbles. Foulk (1937a) has put forward a somewhat similar theory, and pointed out that the bringing together of positively and negatively adsorbed substances in solution in the correct proportions destroys the foam-forming properties of both.

The concentrations at which soluble salts begin to stabilize steam bubbles may be indicated by their threshold carry-over values. Foulk found these to be 4235 p.p.m. for sodium chloride, 3946 p.p.m. for sodium carbonate, and 3151 p.p.m. for sodium hydroxide These values, of course, apply only to the particular conditions of the tests, the effect of pressure and other factors being unknown. It is known, however, that the stabilizing effect of salts having the same kind of adsorption is additive. Compared with the carry-over limits found by Foulk, Vorkauf (1931) gives 10,000 p.p.m. as the commencement of foamability of sodium chloride solutions, but for other salts he finds the foamability concentrations are about the same as the carry-over concentrations of Foulk.

Although the foregoing foamability concentrations have been observed at atmospheric pressure, they are mostly within the region of the total concentrations existing in low-pressure boiler plants. At higher pressures Vorkauf found that the carry-over limit occurred at progressively lower rates of evaporation; it is probable that the foamability concentration is reduced by increasing the saturation pressure; hence the lower concentrations carried in high-pressure boilers do not preclude the possibility of the non-coalescence of the steam bubbles, or foamability.

The question of foaming will be discussed in greater detail later, and it is only mentioned here to emphasize the need for caution in applying the results of experiments with pure or nearly pure water, to the conditions which exist in a boiler. It seems highly probable that in a boiler the steam bubbles do not coalesce so readily as in the experiments of Brooks and Badger, or in those of Behringer. We must consider, then, what happens to a bubble that does not coalesce with others in its passage from the heating surface to the free water level. Such a bubble will, after leaving the heating surface, continue to increase in size from two causes; first, by expansion of the vapor due to consequent reduction in the static head, and second, by growth due to evaporation from the bubble surface.

The expansion of a bubble on account of reduction in pressure can be assumed to take place isothermally, according to Boyle's Law, namely,

$$v_2 = \frac{p_1}{p_2} v_1 \tag{10}$$

At atmospheric pressure the increase in volume is about 30 per cent for each 10-feet difference in level below the free water surface, but in a boiler operating at 400 pounds per square inch pressure the increase in volume is less than 1 per cent for the same change in level. It is evident that in a high-pressure boiler the size of the steam bubbles will not alter appreciably on account of changes in level, even though they originate 30 or 40 feet below the water level, because the relative decrease in total pressure will only be slight.

Each bubble after it leaves the highly superheated layer of liquid in contact with the heating surface, will continue to increase in size by evaporation from its liquid-vapor interface, but at a considerably reduced rate. Jakob and his co-workers found that the coefficient of heat transfer across the bubble surface was fairly uniform throughout the bulk of the water, with a mean value of about 3200 Btu. per square foot per hour per degree F. If the original diameter of a bubble is x, and after a short period of time δt it increases in size by evaporation to a slightly greater diameter $(x + \delta x)$, then its increase in volume is given by

$$\Delta V = \frac{\pi}{2} x^2 \delta x$$

The mean diameter of the bubble during the period δt is $\left(x + \frac{\delta x}{2}\right)$, and the mean surface area from which evaporation takes place is $\pi x(x + \delta x)$.

Let h be the cofficient of heat transfer from liquid to vapor, θ the temperature difference between the liquid and the vapor in the bubble (i.e. the superheating of the bulk of the liquid), l the latent heat of the liquid, v the specific volume of the vapor; then the volume of vapor produced in time δt is

$$\Delta V = \pi x(x + \delta x) \frac{h \theta v}{l} \delta t = \frac{\pi}{2} x^2 \delta x$$

$$\text{Therefore } \frac{\delta x}{\delta t} = \frac{2 h \theta v}{l} \tag{11}$$

Assuming that the pressure remains sensibly uniform during growth, it will be noted that the rate of growth of a steam bubble is independent of its size. Therefore, at the same saturation pressure, both large and small bubbles will increase in diameter at the same rate. Confirmation of this theory is provided by Jakob's tabulated observations of the rate of growth, under atmospheric conditions, of six steam bubbles, whose initial sizes vary from 0.5 cubic millimeters to 83.5 cubic millimeters, which show an almost constant average rate of growth of diameter of 15.5 millimeters per second. The linear relationship of the rate of growth of the diameter is confirmed by two curves (Figure 12), also recorded by Jakob, for the volumes of two other bubbles having initial diameters of 4 millimeters and 1 millimeter, respectively. If the volumes of the bubbles are reduced to equivalent diameters, two straight lines are obtained.

There does not seem to be any reason why the coefficient of heat transfer (h = 3200 Btu. per square feet per hour per degrees F.) found by Jakob for the bulk of the water should vary with the pressure; and on the assumption that it is constant for all pressures the curve in Figure 13 has

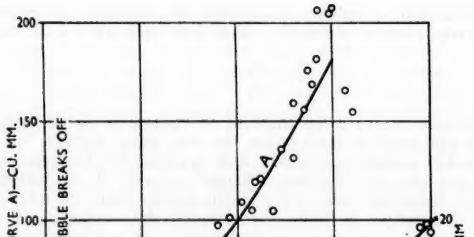


FIGURE 12.—GROWTH OF RISING STEAM BUBBLES AT ATMOSPHERIC PRESSURE (AFTER JAKOB).

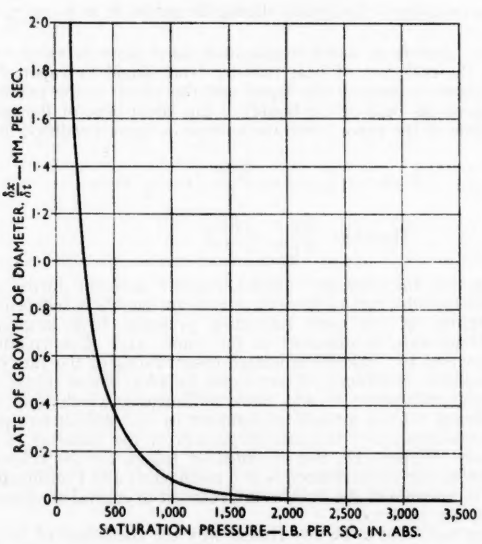


FIGURE 13.—RATE OF GROWTH OF STEAM BUBBLES.
$\delta x/\delta t = 2h\theta v/l = 16.2$ mm. per sec. at atmospheric pressure.

been constructed from equation (11), taking the temperature difference θ from Figure 3 for the superheating of the water. The latent heat l and the specific volume v of the steam are taken from steam tables. It will be noted that at atmospheric pressure the rate of growth is enormous compared with that at boiler operating pressures, but agrees substantially with the value 15.5 millimeters per second deduced from Jakob's observations. Taking the initial bubble size as that obtained from Figure 9 for 45 degrees contact angle, it is equivalent to an increase in diameter of 650 per cent in the first second after release from the heating surface. On the same basis, at 250 pounds per square inch pressure, the increase in bubble diameter in the first second is only 50 per cent; and at 1000 pounds per square inch the increase is only 5 per cent. At higher pressures the rate of growth is almost insignificant, especially when it is remembered that the time taken by the bubbles to migrate from the heating surface to the free water level, even in a large water-tube boiler, is only a matter of two or three seconds.

The actual time taken in migration depends not only on the relative speed of the bubbles with respect to the water, but also on the velocity of circulation of the water in the tubes and the distance traversed. It is not proposed to discuss the circulation in high-pressure boilers, as this has already been dealt with elsewhere (Davis and Timmins 1933), but it is evident that no great error will be introduced into circulation calculations at high pressures by neglecting the velocity of the bubbles relative to the water (Figure 10), in view of the fact that no appreciable increase in the size of the bubbles will occur after they leave the heating surface unless the boiler water is so pure that considerable coalescence takes place.

The Behavior of Steam Bubbles at the Free Water Level.—Bubbles arrive at the free water level in a boiler, partly by the action of convection currents set up by circulation, and partly by the velocity imparted by their own buoyancy. Each bubble as it arrives just below the water level pushes up the surface into a hummock, which instantly drains away on all sides to form a nearly complete hemispherical, dome-shaped, lamina of liquid. Even in pure water the bubbles do not push straight through the surface and burst immediately on arrival, but remain floating at the surface for a fraction of a second; if salts or other stabilizing agents are present, the bubbles may continue to float at the surface for even longer periods before they burst.

The excess pressure inside these evanescent floating bubbles depends on the size of the dome which, being a thin liquid film having two parallel surfaces, conforms to the equation:—*

$$p_1 - p_0 = \frac{4\gamma}{r} \qquad (12)$$

where r is the radius of the nearly hemispherical dome. The water level inside the floating bubble is depressed below the exterior water level by an amount related to the excess pressure ($p_1 - p_0$), the density of the water and the curvature of the surface, which is again a function of the depth. The evaluation of a formula for the shape of this depression leads, therefore, to a very complicated mathematical expression, but in general it conforms to equation (16) in Appendix I. Large bubbles, say greater than 2 centimeters in diameter, have a practically horizontal interior water surface, which is only very slightly depressed below the exterior water level, as indicated in Figure 14a; but in smaller bubbles the excess pressure is greater for the same surface tension, and at about 1 centimeter diameter the interior surface is appreciably curved as in Figure 14b. In still smaller bubbles, say less than 0.5 centimeters in diameter, the water surface becomes nearly hemispherical, as in Figure 14c, but its radius is, of course, only half that of the dome. These observations apply to water under atmospheric conditions.

* See Appendix I, equation (17).

In the bubble dome, the surface tension at the inner surface of the film only balances half the excess pressure, the remainder being transmitted through the liquid in the film to the outer surface. There is thus a squeezing action between the two surfaces, tending to thin the film, the pressure being $\frac{1}{2}(p_1 - p_o)$. This force, assisted by gravity, causes the ultimate failure of the film by thinning at the top of the dome.

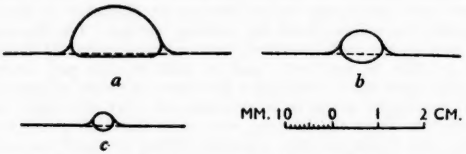


FIGURE 14.—STRUCTURE OF FLOATING BUBBLES.

Some authorities (Stumpner 1936) have attributed the carry-over of water drops in boiler steam to the bursting of the steam bubble domes into fragments; but failure of a liquid film does not take place in this manner. Figure 15, Plate 1, shows frames taken from a high-speed cinematograph film of the rupture of a soap bubble (Edgerton, Germeshausen, and Grier 1936a). It will be noted that even when torn by the impact of a pellet, failure takes place not by rupture into fragments, but by a progressive opening out of the initial perforation. It can be demonstrated that the rate of disruption must be constant, and according to Lawrence (1929) in a soap bubble it takes place at 3200 centimeters per second. Due to the more rapid thinning at the highest part of a bubble dome, disruption will commence at this point, spreading outwardly and presumably without creating drops; nevertheless, when floating bubbles burst, drops of liquid are ejected under certain conditions, and depending on the size of the bubbles producing them, some are projected to a considerable height.

If drops are projected, it is comparatively easy to see them rising and falling above the free water level; they occur not only in boiling, but also in effervescence and with the release of small air bubbles. It is not recorded, however, whether the mechanism of their projection has ever been observed, although a similar action is depicted in the frames shown in Figure 16, Plate 1, reproduced from a high speed cinematograph film (Edgerton and others 1939b) of a drop of milk falling into milk in a container. The drop creates a depression in the milk surrounded by an elevated rim of liquid, similar to the depression and rim in a small floating bubble (Figure 14c), immediately after failure of the dome. Subsequently the rim sinks, and simultaneously the center of the depression rises. Due to the momentum of the inflowing liquid, it throws up a spout from the center of the depression, the tip of which necks off, under the action of surface tension, into separate drops.

If the foregoing interpretation of the mechanism of the disruption of a floating bubble is correct, it would follow that only small bubbles would eject drops, because, as indicated by Figure 14, large bubbles do not depress the inner liquid level sufficiently, in proportion to their size, to cause the center of the depression to rise appreciably on release of the pressure inside the bubble when it disrupts; this is confirmed by observation. Craven and Stuhlman (1931), studying effervescence, found that, under atmospheric conditions, bubbles having diameters up to 1.6 millimeters projected three drops, while those having diameters of 1.6 millimeters—2.0 millimeters projected only two drops. The author's own observations indicate that the first drop may rise to a height of about 10 centimeters but the second only rises 2 centimeters to 3 centimeters above the free water level.

There is no doubt, some relationship connecting the size of the drops and the height to which they are projected with the size of the bubble producing them and the physical properties of the liquid; but the mathematical solution of the problem presents many difficulties. It is suggested that an indication of the underlying relationship can be approximated by considering the vertical force P released at the instant of disruption of a small bubble having a nearly hemispherical depression below the free water level of radius r. In the case of a small hemispherical depression the hydrostatic head is negligible compared with the surface tension forces; therefore if p_1 is the internal pressure and p_0 the external pressure,* then

$$P = \pi r^2 (p_1 - p_0) = 2\pi\gamma r$$

The mass m of water which must be set in motion to replace the depression is $m = \frac{2}{3}\pi\rho r^3$. It will be assumed that force P decreases linearly with time, so the average force applied to the mass m of the liquid is ½P during the time τ which it is acting. No definite value can be assigned to τ, but it must be considerably less than the period of disruption of the dome. In a bubble 2 millimeters diameter the period of complete disruption is less than 1/30,000 second; this application of the force P is, therefore, more in the nature of an impulse. If U is the mean velocity acquired by the mass m under the application of the mean force ½P in time τ then $U = \frac{P\tau}{2m} = \frac{3\gamma\tau}{2\rho r^2}$

It is evident from Figure 16, Plate 1, that the maximum height to which the center of the depression eventually rises is much greater than the mean height of the total mass of moving liquid; it follows then that the height h to which necked-off drops will rise above the free water level will be a function of some considerable multiple of the mean velocity of the total mass m, or $h = \frac{(MU)^2}{2g} = N\left(\frac{\gamma\tau}{\rho r^2}\right)^2$ (13)

Other conditions remaining constant, and considering only surface tension forces, the height h to which the liquid is projected is inversely proportional to the fourth power of the radius of the depression, confirming the observation that small bubbles are much more effective in projecting drops of liquid than large bubbles. This fact has an important bearing on steam contamination.

In a pure liquid the film of liquid forming the bubble dome thins very rapidly, and either disrupts before other bubbles arrive, or is hastened in its collapse by the arrival of other bubbles, since it is inherently unstable and unable to accommodate itself to changes in shape. The rate of thinning does, however, decrease with increase in viscosity of the liquid, and also with increase in the concentration of adsorbed salts. The action of the latter is due to the osmotic pressure, set up between the two surfaces of the film, which opposes the squeezing action caused by the excess vapor pressure inside the bubble dome.

Nevertheless, concentration of salts and increase in viscosity do not endow the bubble film with plasticity; this quality can only be conferred by the presence of finely divided solids. If the bubble film contains a sufficient number of solid particles held together by the liquid, each particle will be capable of a certain degree of movement relative to neighboring particles, the intermediate connecting liquid film acting as a link or adjustable bond

* See Appendix I, equation (16).

between them; the film as a whole is, therefore, more capable of adjusting itself to slight deformation without consequent failure. Very fine particles which form colloid solutions, and which are positively or negatively adsorbed at the bubble film surfaces, produce a highly viscous layer of adsorbed material in the film, giving it both plasticity and stability.

Steam bubbles in which the liquid film is stabilized either by particles of solid matter, or matter in the colloidal state, may remain at the free water level long enough for other bubbles to arrive and collect around them without immediately collapsing or coalescing, and thus form a layer of relatively stable foam. The depth of the foam layer will depend on the rate of thinning of the upper film relative to the rate of arrival of fresh bubbles from below.

Steam Contamination.

In spite of all the precautions taken in modern steam generating plant to prevent steam contamination from entrained moisture and solids, trouble due to this cause is sometimes persistent and recurrent, occurring often most unexpectedly and inexplicably. The importance of preventing steam contamination is evident; it causes loss in thermal efficiency, since the moisture lowers the steam temperature; in large quantities it may also be responsible for direct mechanical damage to the turbine. If salts contained in the moisture are deposited on the turbine blades, the deposits will reduce the mechanical efficiency of the turbine; salt deposits occurring in the superheater tubes (Figure 17, Plate 2) may cause overheating of the metal, culminating in burst tubes. The solids deposited by carry-over are mostly soluble, consisting chiefly of sodium hydroxide, which can be removed by continual washing out with water. In some cases, however, sodium silicate is also carried over and forms an impervious layer of silica, which cannot be so easily removed (Tray 1938).

Mechanical Priming.—Although consideration of mechanical priming caused by faults in design is outside the scope of this paper, it is important to be able to distinguish when steam contamination is due to mechanical priming, and when it is due to other causes of a more physical nature.

A frequent cause of mechanical priming is high-water level, characterized by the entrainment of slugs of water in the steam. High-water level may be due to a variety of causes, but is usually associated with sudden changes in steam demand.

A sudden increase in the demand for steam may result in the phenomenon known as "swelling." If the steam demand momentarily exceeds the corresponding rate of heat adsorption, the boiler pressure drops, and the water is then substantially superheated with respect to the vapor in the migrating steam bubbles. It is clear from equation (11) that the rate of growth of the migrating steam bubbles will be enormously increased, and the resulting increase in volume of the mixture circulating in the boiler will temporarily raise the water level.

A temporary or oscillating high water level may be caused by local reversal of circulation, which takes place more commonly during starting-up, or when the feed is curtailed. If the rate of flow of the water in heated downtake tubes is very nearly the same as the terminal velocity of the steam bubbles tending to rise, then the steam bubbles will be almost stationary relative to the tube, and in time so much steam will collect in the tube that the effective static head of the mixture in the tube may become less than that in the risers, in which event reversal of flow will take place. Again, if the water in a heated downtake tube is normally prevented from evaporating by the entry of cold feed absorbing sensible heat, curtailment of the feed may result in the evolution of steam in the tube, with reduction of the static head; if appreciable, this may again cause reversal of circulation. There is a great deal to be said in favor of the practice of unheated downtake tubes, especially with high pressures.

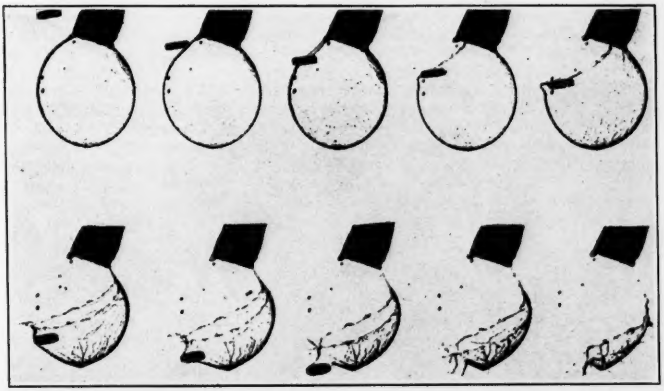


FIGURE 15.—COLLAPSE OF A SOAP BUBBLE AFTER BEING BROKEN BY A SMALL PELLET.
Selected frames from a high-speed motion picture taken at 1200 per second (after Edgerton and others).

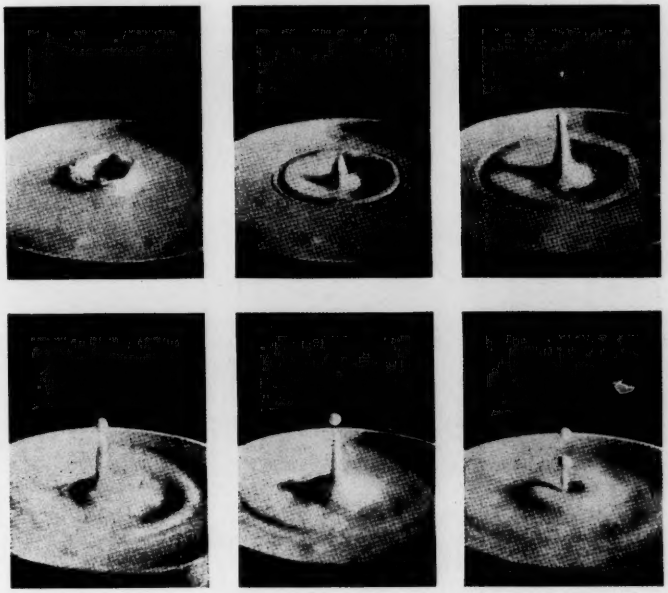


FIGURE 16.—SPOUT AND DROP FORMATION CAUSED BY A DROP OF MILK FALLING INTO MILK.
Selected frames from a high-speed motion picture (after Edgerton and others).

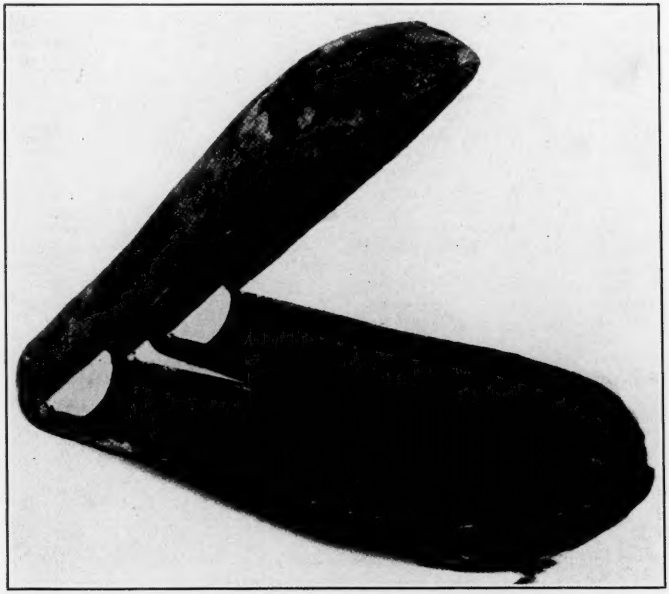
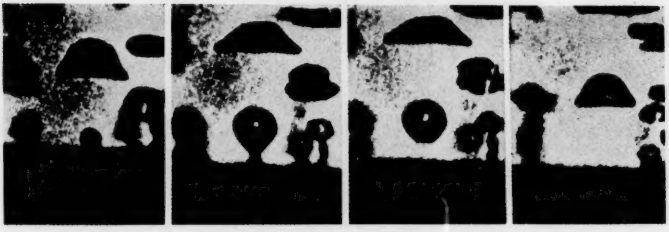


FIGURE 17.—SUPERHEATER TUBE BEND CHOKED BY SALTS DEPOSITED BY CARRY-OVER.



1	12	13	24
0.001 sec.	0.020 sec.	0.022 sec.	0.041 sec.

FIGURE 18.—BUBBLE FORMATION AT A HEATED SURFACE.
Selected frames from a series of 24 high-speed motion pictures (after Jakob).

Foaming.—Although it has been shown that suspended solids can stabilize a foam, some investigators have found that under certain conditions they have little or no effect. Foulk (1936) made laboratory experiments at 250 pounds per square inch and found that much depended on the mode of formation of the solids. For instance, calcium carbonate varied in its effect according to the way in which it was precipitated. One possible explanation of Foulk's experience with calcium carbonate precipitates may be that the physical condition of the precipitates was different in each case; for example, the grain size of a product formed in a precipitation process depends, among other things, upon the hydrogen-ion concentration.

It is well known that some solids in particle form will float on the surface of a liquid even though the liquid has a considerably lower specific gravity than the solid. For a solid to float in this manner, the contact angle must be greater than zero, and it must also be possible to vary the angle by hysteresis; with an absolutely constant angle a slight tilt would cause the particle to sink. Where particles float at the surface, observation reveals that the surface of the liquid at the point of suspension of each particle is bent down to form a dimple or depression, at the bottom of which the particle hangs suspended in the liquid (Figure 19). The evaluation of the forces acting upon a floating particle is difficult, because of the complex shape of the depression in which it is supported. The conditions to be satisfied by the curvature of the surface at any point at a distance h below the horizontal surface of the liquid (Figure 19) is expressed* by the relationship

$$\frac{1}{r_1} + \frac{1}{r_2} = \frac{gh}{\gamma}(\rho - \rho')$$

where r_1 and r_2 are the two principal radii of curvature of the surface at the point in question, ρ is the density of the liquid, γ its surface tension, and ρ' the density of the vapor.

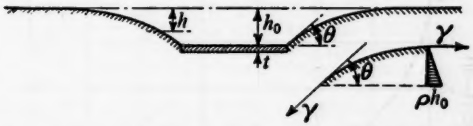


FIGURE 19.—DIAGRAM OF PARTICLE FLOATING AT SURFACE OF A LIQUID.

The only direct method of solving the above equation is to assume that the particle is a disk of very large radius compared with the radius of curvature of the depression in the vertical plane. In most cases this assumption will not err very far from the truth, because, if a number of particles are floating on the surface of the liquid, they tend to collect into diskoid aggregates, due to the unbalanced surface tension forces between them. In this case the disk is mainly supported by the hydrostatic pressure of the liquid, the surface tension merely serving to prevent the liquid from flowing over the disk. If θ' is the contact angle at the depth of h_0 (Figure 19) of the upper surface of the disk below the liquid level, then $\gamma(1 - \cos\theta') = \frac{1}{2} g h_0^2 (\rho - \rho')$

Further, if ρ_1 is the density of the solid, then t, the maximum thickness of the disk compatible with stability (neglecting the vertical component of the surface tension at the edges), is given by the relationship

* See Appendix I, equation (16).

$$\rho' h_o + \rho_1 t = (h_o + t)\rho$$

Hence, substituting for h_o

$$t = 2\sqrt{\frac{\gamma}{g(\rho - \rho')}}\left(\frac{\rho - \rho'}{\rho_1 - \rho}\right)\sin\frac{\theta'}{2}$$

or, substituting the value of γ from equation (3), the maximum thickness t, in millimeters, is given by

$$t = \frac{5.40\,(\rho - \rho')^{2.5}}{\rho_1 - \rho}\sin\frac{\theta'}{2} \qquad (14)$$

where ρ_1, ρ and ρ' are the densities of the solid, the liquid, and its vapor respectively, and θ' is the contact angle between the solid and the liquid.

The foam-stabilizing power of precipitated particles of boiler sludge, which are mechanically suspended and circulated in the boiler water, will depend on whether the particles are smaller than the limiting size t given by equation (14), which in turn is a function of the operating pressure, the density of the particles, and the contact angle. "Wettable" solids having zero contact angle will not float at the surface nor provide a stabilizing scum for foam formation. Some substances will, however, float when scattered on water in a dry condition, but due to the hysteresis of the contact angle, will no longer float after being submerged; the lesser contact angle is, therefore, the criterion for boiler water precipitates. These precipitates consist mainly of calcium carbonate or calcium phosphates, depending on the chemical treatment. Taking the carbonate as an example, although its contact angle as precipitated in a boiler is not known (it may possibly be zero after prolonged boiling), that for calcite, a substance of somewhat similar composition, is given by Sulman

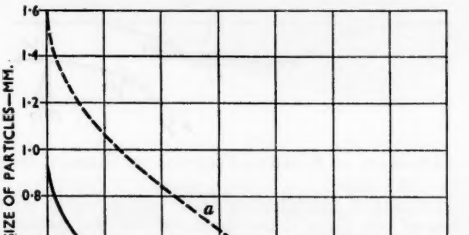


FIGURE 20.—LIMITING THICKNESS OF FLOATING PARTICLES OF CALCIUM CARBONATE.

$$t = \frac{5.40\,(\rho - \rho')^{2.5}}{\rho_1 - \rho}\sin\frac{\theta'}{2}$$

(Taggart 1927) at the lower limit as $\theta' = 39.6$ degrees. The density ρ_1 of calcium carbonate is 2.8. A curve showing the probable limiting size for floating calcium carbonate sludge particles is given in Figure 20, calculated by means of equation (14). According to Figure 20, the largest sludge particles capable of floating at the free water surface at ordinary boiler pressures are of the order of 0.5 millimeters in thickness.

The question arises, How are the particles conveyed from the interior of the boiler water to the free water surface? The steam bubbles in moving through the water collect widely dispersed particles, in the same way that air bubbles do in the froth flotation of mineral ores, whereby selected minerals are separated by this process, according to their contact angles. Providing the particles have a finite contact angle, that is, they are not "wettable," they will adhere to the bubbles by the same mechanism that enables them to float at the free water surface. As each bubble collapses at the free water level, it deposits its load of particles, which ultimately form a scum; when foaming takes place this scum is deposited just above the water level on the walls of the boiler drum, and sometimes in the steam outlets.

The mechanics of flotation has been dealt with in detail by Wark (1933); its analysis is very complicated, but it is evident that the contact angle is in this case the limiting factor, since from considerations of buoyancy alone the steam bubbles could transport particles nearly twice the size limited by floatability.

Particles of calcium carbonate sludge which are larger than the limiting size given by curve a in Figure 20 will not contribute to form a scum, and will have no influence on the foaming properties of the boiler water. It has been observed that coarse granular material generally has no effect on foaming, but the finer the precipitate the more likely it is to cause foaming.

The solids, in precipitating out from the boiler water, must pass through a stage when the size of the particles is within the colloidal region, which stabilizes the bubble films; but unless the boiler water contains a peptizing agent, the particles will continue to grow in size and may even coalesce or flocculate into a true precipitate, in which case the colloidal stage will only be transient and will have very little influence upon the steaming properties of the water. Unfortunately boiler water always contains a large proportion of caustic alkali (NaOH), which is a very effective peptizing agent.

That sodium hydroxide, and possibly also sodium phosphate, have some effect on the foaming qualities of boiler water has already been observed, but lack of understanding of their true function has led to confusion of the evidence. It was demonstrated, for instance, that a solution of sodium hydroxide up to 50,000 p.p.m. did not foam. Later experiments showed that no foaming occurred with high concentration of sodium salts unless suspended matter was present in the water (Powell 1937). No doubt some of the suspended matter was in this case converted to the colloidal state. According to Niehaus (Stumpner 1936), colloids in boiler water lead to persistent foaming, a degree of dispersion of the order of 0.1μ diameter having the maximum effect.

The proportion of sodium hydroxide to total solids has been found to be an important factor in foaming, and Tray (1938) gives a curve showing the effect of the ratio of sodium hydroxide to total dissolved solids in boiler water (Figure 21). The extent of the foaming is indicated by the amount of solids carried over in terms of the conductance of the steam. To quote from Tray, "A study of these test results shows that the total dissolved solids exert less influence on carry-over than does alkalinity. An increase in total solids by itself does not result in more carry-over unless such increase is accompanied by an increase in the ratio of alkalinity to total solids. It will be noted that the curve in Figure 21 changes slope when the alkalinity is approximately 17 per cent of the total solids and a further increase in this ratio results in a more rapid increase in conductance of the steam."

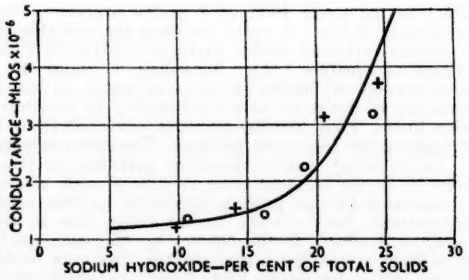


FIGURE 21.—EFFECT OF ALKALINITY ON STEAM QUALITY (AFTER TRAY).

+ High make-up plants.
O Low make-up plants.

Caustic alkali also decomposes organic materials, such as oil, sewage, and humic matter into colloidal decomposition products. Boiler water contaminated in this manner is, therefore, very liable to foam. The source of contamination can often be traced to the use of impure cooling water, and its introduction into the boiler circuit through condenser leakage.

Evidently, methods to be adopted to prevent foaming must in each case depend on the specific cause, which is not often easy to elucidate. As a general rule, it is advisable, if foaming occurs, to reduce the concentration of the dissolved solids in the boiler water, and at the same time the alkalinity should not be allowed to exceed 15 per cent of the total dissolved solids then remaining. If a continuous blow-down is needed to reduce the concentration, it should be fitted to the drum at which maximum effective concentration occurs; for instance, in a three-drum boiler this would be at the upper front drum. The precipitated solids, which settle out in the lower drums and headers, obviously cannot contribute to the foamability of the water, and need only be cleared out intermittently.

The presence in the boiler water of substances having coagulating properties, such as aluminum hydrate and tannins, has been observed to reduce foaming. Sodium aluminate has also been successfully used as a coagulant. It is also recorded (Matthews 1938) that the addition of castor oil to boiler water can prevent foaming. The action of castor oil is obscure; it may have some coagulating action, or it may become adsorbed on the surfaces of the suspended particles resulting in reduction of the contact angle.

Where organic matter is the original cause of foaming, Tray recommends the addition of a small quantity of activated carbon, fed to the boiler internally, together with sufficient iron as hydrous ferrous oxide to coagulate the suspended carbon. It is claimed that the organic matter can be completely absorbed by the carbon, while the iron floc prevents any increase in suspended solids.

Carry-over. As already described, small steam bubbles, disrupting at the water level, project minute drops of water into the steam space. The height to which the drops rise above the water level depends not only upon their velocity of projection, but also on the resistance of the rising vapor and its velocity. The influence of these factors on the height of projection is expressed by equation (19), Appendix II. The only term in the formula to which a definite value can be given is the velocity of the steam c. Until experimental evidence is available, the determination of both p, the velocity of projection, and q, the terminal velocity of the drops, must

remain somewhat hypothetical. From equation (13), it is to be expected that the velocity of projection will be proportional to $\gamma/\rho r^2$ which, when investigated, leads to the conclusion that the velocity of projection is very nearly the same at all saturation pressures; further, from observation of the disruption of small air bubbles, in still atmospheric conditions, the height of projection in this case is about 10 centimeters. Neglecting the air resistance, $p = \sqrt{2gH} = 140$ centimeters per second (approximately). It will be assumed then, tentatively, that $p = 140$ centimeters per second at all pressures.

It has also been shown that the vertical force P released at the instant of disruption of the bubble is equal to $2\pi\gamma r$, where r is the radius of the depression. The energy expended is, therefore, $Pr/2 = \pi\gamma r^2$. If the whole of this energy is communicated to a single drop of weight W, which rises to a maximum height H, then $W = \pi\gamma r^2/gH$. From this relationship it can be concluded that if a bubble 2 millimeters in diameter projects a drop to a height of 10 centimeters, the weight of this single drop must be less than 0.2 milligram; at least two drops can be observed, although they do not both rise to the same height. The weight of the first drop, therefore, is probably less than 0.1 milligram, and its diameter something less than 0.6 millimeters. At higher saturation pressures, assuming p to be constant, H will also be constant in an atmosphere without resistance, and the weight of the drops will be proportional to γr^2, from which it follows that, approximately,

$$d_0 \propto \left(\frac{\gamma d^2}{\rho}\right)^{\frac{1}{3}}$$

where d is the mean diameter of the bubble, and d_0 the diameter of the drop, both in millimeters. From the conjectured size of the drops, it is now possible to estimate their terminal velocity q from equation (18), Appendix II. In this manner the curves of Figure 22 have been prepared to show the effect of the velocity of the released steam on the height of projection of the water drops, at three different pressures.

At low pressures an increase in the rate of steam release has an appreciable effect upon the height to which the drops rise; this is confirmed by Vorkauf's conclusions that at low pressures the height of the steam space has considerable influence on the wetness of the steam. Obviously, if the drops are projected so high that they enter or approach the steam outlets (or any other region where the vertical component of the steam velocity exceeds the terminal velocity of the drops), the drops will be entrained with the steam. Vorkauf concluded from his experiments, carried out at comparatively low pressures, that both the height of the steam space and the area of steam release were effective in controlling the wetness of the steam, and deduced therefrom that the steam space volume was the important factor. Actually, the area of release and the height are two separate factors, one of which—the height—becomes decreasingly important as the saturation pressure increases. This is evident from the curves (Figure 22) for 600 pounds per square inch and 1400 pounds per square inch; at the latter pressure the water drops probably do not rise more than about 2 inches above the water level; but carry-over will occur much more suddenly and at a much lower steam velocity. It was discovered by Vorkauf, and since frequently observed in practice, that true carry-over occurs suddenly, when the evaporation has reached a certain definite rating; the nature of the curves in Figure 22 indicate that this is to be expected.

The relationship between the probable carry-over velocity of the steam and the operating pressure is shown in Figure 23. The curve of permissible

rating allows for a factor of safety of 4 on the calculated carry-over velocity, to allow for inaccuracies in the assumptions made and variation in the local, as compared with the calculated, average rate of steam release.

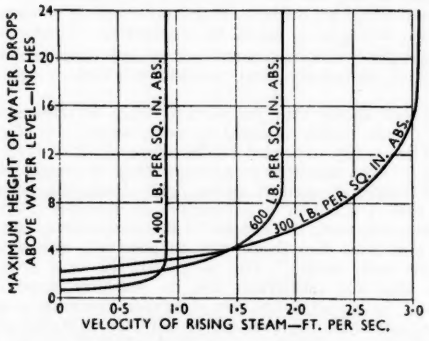


FIGURE 22.—PROBABLE HEIGHT OF PROJECTION OF WATER DROPS ABOVE WATER LEVEL.

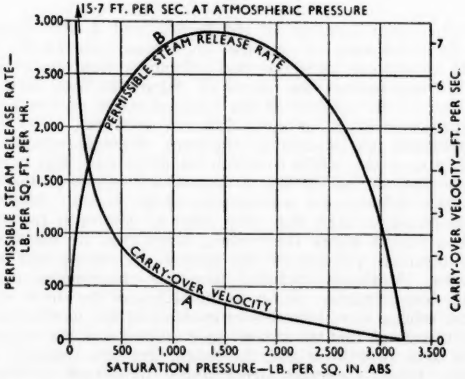


FIGURE 23.—ESTIMATED CARRY-OVER VELOCITY OF STEAM, AND RECOMMENDED PERMISSIBLE STEAM RELEASE RATE.

The most objectionable feature of carry-over is the deposition of the dissolved salts from the boiler water, when the drops projected by the disrupting steam bubbles, which are carried over with the steam, eventually evaporate. The water drops partially evaporate during superheating of the steam, some of the solids then forming deposits in the superheater tubes (see Figure 17, Plate 2), while the remainder may pass on and cause deposits on the turbine blades.

A comparatively small amount of carry-over can cause appreciable trouble at the turbine; with boiler water containing only a few hundred parts per million of solids, several pounds of deposit may form during a day's run, even if there is only 0.5 per cent moisture in the steam. The chemical which causes the main difficulty is sodium hydroxide, the concentration of which varies with the temperature and steam pressure. According to the curves reproduced in Figure 24 from Straub's (1936) investigations, sodium hydroxide solution at the steam temperatures and pressures used in modern plants may contain from 10 to 20 per cent moisture, resulting in a semi-fluid sticky solution, which will adhere to metal surfaces, accompanied by some of the

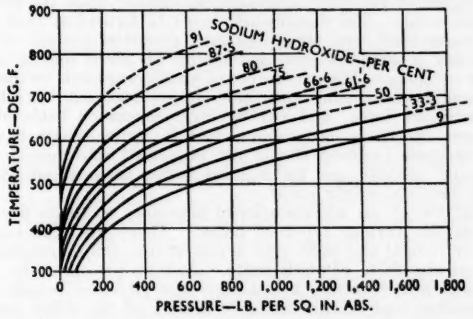


FIGURE 24.—CONCENTRATION OF SODIUM HYDROXIDE, IN RELATION TO TEMPERATURE AND STEAM PRESSURE (AFTER STRAUB).

salt residues having high fusion temperatures, such as sodium chloride, sodium carbonate, and sodium sulphate. It is the adhesive property of the sodium hydroxide which previously led to the belief in the so-called selective carry-over of solids, because the composition of the deposits bears no apparent relation to the ratios in which the materials occur in the boiler water; the proportion of sodium hydroxide is usually excessive.

Straub suggested that the adhesion of the sodium hydroxide might be prevented by the addition to the boiler water of more solids of a powdery nature, or alternatively by adding certain organic compounds to the boiler water; either of these methods, however, may lead to further trouble by causing foaming. An alternative means is to remove the salts from the moisture, or to remove the moisture from the steam.

The position of the steam inlets to the drum has been shown to have a great influence on the steam quality. Vorkauf found that reduction in the wetness occurred if the steam inlets of riser tubes discharged above the water level, instead of below it; and in cases of foaming, it had the added effect of depressing the foam layer. The advantages of this construction may depend to some extent on the physical qualities of the steam and water mixture discharging from the riser tube, which again may vary with the rate of circulation and the chemical composition of the boiler water. Only those tubes which are expected to discharge a mixture containing a very large proportion of steam should have their outlets above the water level.

From Brooks and Badger's (1937) experiments it can be inferred that if the proportion of steam in the mixture leaving a riser tube is relatively small, then the mixture will discharge above the water level as a froth, and if

there are sufficient dissolved salts in the water, there is no doubt that a froth discharge will tend to persist with still greater proportions of steam in the mixture. If, on the other hand, the boiler water has a low concentration of solids, as the proportion of steam increases, the bubbles will coalesce and the discharge will take place in alternate slugs of water and vapor.

To produce reasonably dry steam, control must commence with the separation of the steam from the water in the boiler drums. It has been shown experimentally by Vorkauf that proper control at this stage can exert considerable improvement in the quality of the steam. The arrangement of the internal baffles, fittings, and boiler tubes entering the drums should, therefore, be designed in the most advantageous manner from the point of view of drying the steam. The steam outlet must be located as high as possible above the water level, and above the least disturbed portion of the water surface; further it should be protected against the direct ingress of projected water drops or spray. The carry-over of small drops can be diminished by allowing them the maximum opportunity to separate out: (a) by gravity; (b) by centrifugal force; and (c) by impingement on baffle plates. The drying process should be progressive, that is, steam which has once had the initial moisture removed should not be passed subsequently through a zone of greater wetness, and the dividing baffles between subsequent steam spaces must have steamtight joints.

The application of the aforementioned principles to steam drying can be as diverse as the various types of boiler. There are also many special designs of centrifugal and baffle type dryers in use, and descriptions of these have appeared elsewhere (Powell 1937). At high pressures the centrifugal method of separation becomes increasingly less effective, because the water drops are smaller, and the difference in density of the steam and water is

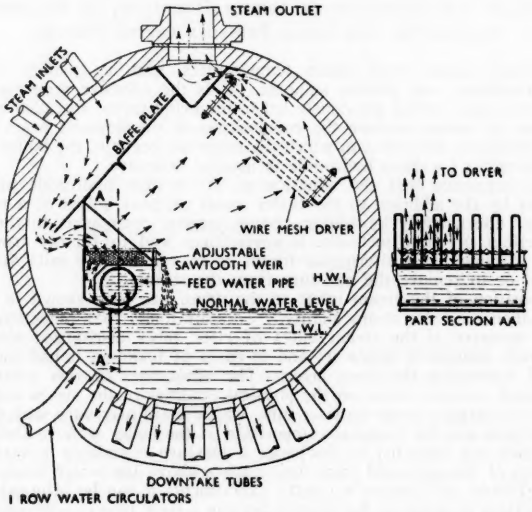


FIGURE 25.—DIAGRAMMATIC ARRANGEMENT OF STEAM WASHER.

less (Figure 1). Although other methods of separation have been suggested, such as electric glow discharge and ultrasonics, the alternative principle has lately been adopted, more especially in America (Edison Electric Institute 1938), of washing the steam by the incoming feed water. Any moisture consequently entrained in the steam by washing then has a much lower concentration of solids than the boiler water.

Steam washing is achieved either by passing the steam through baffles wetted by the feed water, or by bubbling the steam through the feed water (Figure 25), or, in another type, by spraying the feed water into the steam; subsequently the steam is passed through some form of dryer to remove the excess of moisture, the whole process taking place in the main steam and water drum. The satisfactory performance of a steam washer would seem to depend on the relative positions of the steam and the water inlets to the drum, and on the control of the direction of flow of the steam and the water. The application of steam washers in this country has so far been rather limited, and not always successful.

Conclusions.

Theoretically, the physical properties of the boiler water may profoundly affect its steaming qualities, and changes in these qualities must be expected at higher operating pressures. Normally, evaporation at the heating surface takes place by nuclear boiling, and with increase in the operating pressure the size of the bubbles generated becomes smaller. During migration from the heating surface to the free water level, the bubbles grow considerably in size at low pressures, but only slightly at high pressures.

Due to these causes the average size of the steam bubbles reaching the free water level at low pressures is much greater than those arriving under the conditions of high operating pressures, unless the bubbles formed at higher pressures coalesce into larger units. The size of the bubbles reaching the surface will not be uniform, however, owing to differences in the qualities and slopes of the generating surfaces, rates of circulation in the boiler tubes, and periods of growth during migration. Nevertheless, a preponderance of large bubbles reduces the probability of water drops being ejected into the steam space; it is the small bubbles which are more likely to contribute to the carry-over of moisture in the steam.

The more dissolved solids there are in the water, the less is the tendency for the steam bubbles to coalesce; but if the concentration of dissolved solids is kept low, the small bubbles more readily coalesce and form larger units. These large bubble units are not so likely to project water drops into the steam space upon disruption at the water level; they also induce a faster rate of circulation. It is now clear why, in order to preserve a reasonable purity of steam, a concentration of solids in the boiler water, which may prove satisfactory at low pressures, cannot be tolerated at high pressures, because it stabilizes the undesirable small bubbles which are formed at high pressures.

Although the dissolved solids in boiler water do not appreciably affect the surface tension, they do influence the stabilization of the bubbles by adsorption at the steam and water bubble interfaces. Persistent foam does not result, however, from this partial stabilization by the dissolved salts, but is due to the presence of suspended matter, principally in the colloidal state. It is to minimize the stabilization of foam by colloids that the proportion of caustic alkalinity must be kept low and organic matter practically excluded from boiler water. To what extent foaming is directly influenced by increase in the saturation pressure is uncertain.

Clearly, in designing a high-pressure boiler, much may be accomplished towards ensuring purity of the steam by careful design of the internal baffles and fittings. When the boiler goes into operation an additional factor must

be considered; the steaming properties of the boiler water depend almost entirely on the nature of the feed water and the careful control of its chemical treatment, not only in relation to the purely chemical changes brought about, but—equally important—to the physical and chemico-physical changes. Existing chemical treatments to protect a boiler against scale, corrosion and embrittlement are not as a rule designed to prevent steam contamination—they may even be the cause of it. Looking to the future, chemical treatment should be aimed at, which while still adequately protecting the boiler metal also ensures purity of the steam. This will only be achieved by giving due consideration to the chemico-physical reactions of the chemical treatment of the boiler water upon the physical processes of steam generation.

Appendix I.

Definitions.

Surface Tension. The molecules of a liquid in contact with a gas tend to move in towards the interior of the liquid. As a result, the liquid behaves as if surrounded by an elastic skin, with a tendency to contract. The tension in this apparent skin is measurable; it is defined as the surface tension, and is measured in dynes per centimeter of length.

It follows from the theory of molecular attraction that the surface tension will be a function of the density of the surface layer and will, therefore, vary both with temperature and the nature of the gas or vapor in contact. For a liquid in contact with its own vapor, Macleod (1923) gives the following simple rule

$$\gamma = K(\rho - \rho')^4 \qquad (15)$$

where γ is the surface tension, ρ and ρ' are the densities of the liquid and its vapor respectively, and K is a constant independent of temperature.

The equation connecting the difference in the pressures on the two sides of a liquid-gas interface with its curvature is

$$\frac{1}{r_1} + \frac{1}{r_2} = \frac{p_1 - p_0}{\gamma} \qquad (16)$$

where r_1 and r_2 are the principal radii of curvature at the point under consideration, and p_1 and p_0 are the pressures on the inside and outside of the surface respectively.

In the case of a thin liquid film there are two parallel surfaces, and the equation connecting the difference in pressure on the two sides of the film with its curvature is

$$\frac{1}{r_1} + \frac{1}{r_2} = \frac{p_1 - p_0}{2\gamma} \qquad (17)$$

where p_1 and p_0 are the pressures on the inside and outside of the film, respectively.

Adsorption.—Some substances when dissolved in water raise the surface tension; others lower it. The solutions of most inorganic salts raise the surface tension of water. This is due to the process of adsorption. The surface phase of a solution is a film or lamina one molecule thick with definite and measurable properties differing from those of the solution in bulk. If increasing the concentration decreases the surface tension, then the dissolved substance will collect in the surface layer; this is called positive adsorption. If, on the contrary, the surface tension increases with concentration, then the surface layer will be more dilute and the adsorption is said to be negative.

Contact Angle.—When a liquid does not freely spread over a solid and wet it, the surface of the liquid joins the solid at an angle. The angle between the surface of the liquid and the solid-liquid interface is within limits fairly definite and constant for any particular combination of solid and liquid and is called the contact angle. The contact angle is, however, greater when the liquid is approaching the dry surface than when it is receding from the previously wetted surface. This effect is sometimes referred to as hysteresis of the contact angle and is thought to be a manifestation of friction.

Dispersion of Solids in Liquids.—When a liquid is adsorbed by a solid it tends to peptize or disintegrate it, and in some cases it will do so, if the attraction of the molecules of the liquid for those of the solid exceeds the attraction of the molecules of the solid for each other. At higher temperatures the molecular activity is greater, and the peptizing action increases. If the dispersion is profound enough, the solute may go into true or crystalloidal solution, but if the molecules of the solute cling to each other to a considerable extent they may form groups, of colloidal dimensions, and a colloidal "solution" results. The particles in a colloidal solution range between $1\mu\mu$ and 0.1μ, while those in a true crystalloidal solution are less than $1\mu\mu$ ($1\mu = 0.001$ mm.).

Still larger particles than those in a colloidal solution between the range 0.1μ to 1μ, although not possessing all the properties of a true colloid, yet will form a permanent suspension, being maintained in continuous motion by Brownian movement arising out of successive molecular collisions. Particles above 1μ usually form precipitates. Suspensions and precipitates may also be formed by the action of a flocculating agent, which has the converse action of a peptizing agent, causing the aggregation or agglomeration of finer particles.

For a more detailed discussion of the physical properties of surfaces reference may be made to the works of Alexander (1929), Edser (1922), Newman and Searle (1933), Schmaltz (1936), Wark (1933), and others.

Appendix II.

Height of Projection of Drops from Disrupting Vapor Bubbles.

The problem is to find the maximum height attained by a liquid drop, which is projected vertically upwards by the disruption of a small vapor bubble at the liquid level. It will be assumed that p, the velocity of projection of the drop, is greater than c, the velocity with which the vapor rises from the surface; further, that velocity c remains constant throughout the height of projection of the drop.

All quantities having an upward direction will be denoted positive and those having a downward direction negative. If v is the velocity of the drop relative to the vapor, then at the instant of projection ($t = 0$),

$$v = p - c$$

It can be assumed that the resistance R of the vapor to the motion of the drop will be turbulent, and therefore, according to Newton's Law

$$R = Kv^2$$

When v is positive, R is negative, and during the first stage of the path of the drop upwards, the resultant force Y acting on the drop is

$$Y = -R - W$$

where W is the apparent weight of the drop in the vapor.

The relative velocity v remains positive until the drop reaches the point where $v = 0$; thereafter v becomes negative and R positive, and the resultant force acting on the drop during the second stage is

$$Y = R - W$$

The drop reaches its maximum height when $c - v = 0$.

To simplify the mathematical analysis, the motion of the drop will be considered relative to a system of coordinates, which is moving with the vapor at the same velocity c vertically upwards relative to the liquid level, with which it will be assumed that the origin coincides at the instant $t = 0$. At any other time t', the origin of the coordinate system will be at a height y' above the liquid level corresponding to

$$y' = ct'$$

The acceleration of the drop at any instant t will be dv/dt; hence equating the forces acting on the drop during the first stage

$$Y = \frac{W}{g}\frac{dv}{dt} = -Kv^2 - W$$

And during the second stage

$$Y = -\frac{W}{g}\frac{dv}{dt} = Kv^2 - W$$

Put $q^2 = W/K$; the equations of motion then become:—

During the first stage $$\frac{dv}{dt} = -g\left(1 + \frac{v^2}{q^2}\right)$$

During the second stage $$\frac{dv}{dt} = g\left(1 - \frac{v^2}{q^2}\right)$$

If y is the height of the drop, at the same instant t, above the origin of the moving coordinate system,

$$v = \frac{dy}{dt}$$

Therefore $$\frac{dv}{dt} = v\frac{dv}{dy}$$

and the equations of motion can be transposed to the following forms:—

For the first stage $$v\frac{dv}{dy} = -g\left(1 + \frac{v^2}{q^2}\right)$$

For the second stage $$v\frac{dv}{dy} = g\left(1 - \frac{v^2}{q^2}\right)$$

If T is the total time from projection for the drop to attain its maximum height above the liquid level and h is the corresponding height of the drop above the moving coordinate system at the end of the same interval of time, then the limits of integration for this condition are:—

$$t = 0 \quad y = 0 \quad v = p - c$$

and $$t = T \quad y = h \quad v = c$$

Hence $$g\int_0^T dt = -\int_{p-c}^{0} \frac{dv}{1+\frac{v^2}{q^2}} + \int_0^c \frac{dv}{1-\frac{v^2}{q^2}}$$

and $$g\int_0^h dy = -\int_{p-c}^{0} \frac{v\,dv}{1+\frac{v^2}{q^2}} + \int_0^c \frac{v\,dv}{1-\frac{v^2}{q^2}}$$

which give upon integration

$$T = \frac{q}{2g}\left[2\tan^{-1}\left(\frac{p-c}{q}\right) + ln\left(\frac{q+c}{q-c}\right)\right]$$

$$h = \frac{q^2}{2g}\left[ln\left\{1+\frac{(p-c)^2}{q^2}\right\} - ln\left(1-\frac{c^2}{q^2}\right)\right]$$

The maximum height H attained by the drop above the liquid level is given by $H = h + cT$, that is

$$H = \frac{q^2}{2g}\left[ln\left\{1+\frac{(p-c)^2}{q^2}\right\} - ln\left(1-\frac{c^2}{q^2}\right)\right] + \frac{cq}{2g}\left[2\tan^{-1}\left(\frac{p-c}{q}\right) + ln\left(\frac{q+c}{q-c}\right)\right]$$

In a still vapor, $c = 0$, and

$$H = \frac{q^2}{2g} ln\left(1+\frac{p^2}{q^2}\right)$$

Further, if the resistance of the vapor is negligible, $K = 0$; i.e. $(1/q^2) = 0$, and from the expansion of $ln\left(1+\frac{p^2}{q^2}\right)$ for this case

$$H = \frac{p^2}{2g}$$

When $c = q$, $H = \infty$; that is, the drop will never return, hence q is the terminal or carry-over velocity for the drop.

If the drop is spherical, of diameter d_o,

$$W = (\rho - \rho')\frac{\pi d_o^3}{6}$$

and $$K = \frac{N\pi d_o^2}{4}\frac{\rho'}{2g}$$

hence $$q^2 = \frac{W}{K} = \frac{4g}{3N}\left(\frac{\rho-\rho'}{\rho'}\right)d_o$$

where ρ and ρ' are the densities of the liquid and the vapor respectively.

Putting the numerical constant $N = 0.858$, this agrees with Rittinger's formula for turbulent motion of a sphere (Martin 1928), giving the terminal velocity of the drop

$$q = \sqrt{\frac{W}{K}} = 12.35\sqrt{\frac{\rho - \rho'}{\rho'}}\, d_o \qquad (18)$$

where d_o is the diameter of the drop in millimeters, and q is in centimeters per second.

Converting to common logarithms, and putting the tangent of the angle in degrees, the formula for H can be expressed in the more practical form

$$H = \frac{q^2}{853}\left[\log_{10}\left\{1 + \frac{(p-c)^2}{q^2}\right\} - \log_{10}\left(1 - \frac{c^2}{q^2}\right)\right]$$

$$+ \frac{cq}{853}\left[\frac{\tan^{-1}\left(\frac{p-c}{q}\right)}{66} + \log_{10}\left(\frac{q+c}{q-c}\right)\right] . \qquad (19)$$

where H is the maximum height, in centimeters, attained by the drop above the liquid level, q the terminal velocity of the drop, in centimeters per second, as defined above (equation (18)), c the velocity of the vapor rising from the liquid, in centimeters per second, and p the velocity of projection of the drop from the liquid, also in centimeters per second.

Appendix III.

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DISCUSSION.

MR. R. E. ZOLLER observed that the author, commenting on Figures 2 and 3, stated that Figure 3 closely resembled Figure 2. It was difficult to see why the two figures were compared in that way by the author. When he himself read Jakob's paper, the first thing he did was to attempt to express the superheat in terms of the difference in pressure between the inside and outside of a bubble; an interesting curve was thus obtained. At atmospheric pressure, the difference in pressure between the inside and out-

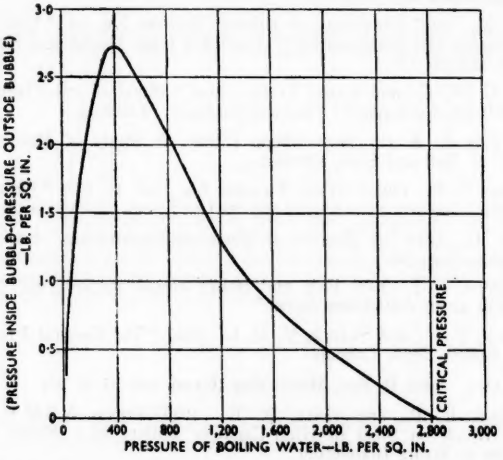

FIGURE 26.—PRESSURE DIFFERENCES BETWEEN INSIDE AND OUTSIDE OF STEAM BUBBLES.

side of the bubble was about 0.33 pound per square inch; and it rose to a maximum of about 2¾ pounds per square inch at about 500 pounds per square inch pressure and then dropped to zero (Figure 26). The ordinates of Figure 26 were expressed in pounds per square inch, and with that essential difference between Figure 26 and Figure 3, he did not arrive at the same inference as the author. The author's comparison between Figures 2 and 3 was merely misleading.

The author stated that if an absolutely plane surface could be obtained it would be possible to achieve nuclear boiling only if there were infinite superheat. Even granted a perfectly smooth surface, infinite superheat could never be obtained, for, as Figure 26 showed, when the critical pressure was reached the difference in pressure between the inside and outside of the bubble was zero; thus the water would pass into steam and the film would start boiling at once.

The author referred to the coefficient of heat transfer between the steam in the bubble and the water, and gave its value as 3200 Btu. per square foot per hour per degree F. The author saw no reason why it should vary with the pressure, and assumed that it was constant. In Jakob's paper several values were given, varying between 1600 and 5600 Btu. per square foot per

hour per degree F. It was admittedly very difficult to measure the heat transfer, but he himself thought that it would vary considerably with the pressure. He regarded this transfer rather as a convection heat transfer due to the relative velocity between the water and the bubble; Figure 11, for instance, showed the bubble velocities at different pressures. At high pressures there was a lower relative velocity between the steam bubble and the water, but there was a higher viscosity and specific heat and a smaller bubble size, and the latter factors tended to increase the heat transfer.

He referred to a bubble of the type shown in Figure 14c, having a diameter of about 0.5 centimeters, the size given on the attached scale. If the steam rose from the surface at about 0.5 feet per second at 500 pounds per square inch, about 30 bubbles would rise through 1 square centimeter of surface every second. Probably the bubbles lasted longer than $\frac{1}{30}$ second, so there would be more than one bubble on a given square centimeter of surface at a time. The point was that the bubbles would be touching and distorting each other. If the bubbles were in close proximity to each other, the effect shown in Figure 16, Plate 1, would not be obtained, because the water would flow into the depression with different velocities from each direction.

Figure 23 gave the recommended permissible steam release rate. He had referred to a number of tests and had found that American practice was in agreement; nearly all the boilers tested were "dry steamers," both land and marine types, and the release rates followed curve B very well. Probably the author based his choice of a factor of safety of 4 on many data, and if he could give more information on that point, it would increase the practical value of the paper.

Mr. G. W. Hewson remarked that the paper was a useful contribution towards emphasizing the great importance of correct design as well as correct operation of boiler plant. The modern high-pressure boiler was not merely an enlarged edition of a low-pressure boiler, made by substituting materials which would withstand higher pressures and temperatures. The author had presented a strong case for careful revision of design by frankly including an illustration of a choked superheater tube (Figure 17, Plate 2). In addition to the choking—with subsequent burning and bursting—of such a tube, there was also the danger of corrosion in tubes, when the boiler was idle, caused by water lodging in the bends and absorbing oxygen and carbon dioxide from the atmosphere. Every part of a boiler system should therefore be made so that it could be completely emptied when it was laid off or "opened up."

The paper was a good indicator of what not to do, as well as of what ought to be done. Particularly in high-pressure boilers the solids in the water should be kept to a minimum, and he would recommend much lower figures than those cited. In two installations at least with which he himself was connected, the solids were kept below 300 p.p.m., and not merely below thousands of parts per million; and in one case the boilers had been working for twelve years without any serious trouble, and in the other case for seven years. The figures cited in the paper were much higher than those usually found in successful modern installations. They were based, he feared, on the practice, long observed with low-pressure boilers, of "doping" with chemicals in excess of the amounts absolutely necessary, particularly in plants where the steam, after doing its work, was condensed and returned as feed to the boilers. That practice was based on the assumption that condensed steam was acid in nature; Major Gregson had emphasized that point. The assumption was based upon rather immature chemical tests; samples were collected in an open vessel, where they immediately absorbed carbon dioxide from the atmosphere, and therefore gave an acid reaction. If the sample was collected, as were samples for the oxygen test, out of contact with the

atmosphere, it was found that the water was of a slightly alkaline nature. The hydrogen-ion concentration p_H value was on the alkaline side of 7.

Regarding steam contamination, by keeping to lower contents of solids than the figures mentioned in the paper, blowing-down when the solids reached the 300 p.p.m. mark, and working with a make-up of under 3 per cent, steam contamination had been practically eliminated in the plants to which he had referred, and there was very little trouble from foam.

Reference had been made to finding out the cause of contamination of feed water or the return water to the boilers. A tester was already in use which indicated whether there was any increase of solids in the water, but not whether it was from carry-over from the boilers or leakage from the condensers. The London Power Company had recently been in contact with some instrument makers in an attempt to devise an instrument for recording the hydrogen-ion concentration of the water. If such an instrument could be perfected and used in conjunction with the tester just mentioned, the cause of feed water contamination could at once be traced by observations of the recording charts of both instruments.

Mr. R. F. Davis wrote in reply that the main purpose of the paper had, admittedly, been to show how the higher pressures now employed might affect steam contamination in large water-tube boilers. Special difficulties had become manifest in this domain, though not to any great extent in this country, as yet. It was to be expected, however, that more and more high-pressure units would come into operation in this country, and it was, therefore, important to anticipate results. Some of his interpretations of the phenomena of steam generation could, nevertheless, be applied even to the low pressures existing in evaporators. It required no great perspicacity to decide to what extent a particular portion of the paper was applicable to other conditions of pressure, or other types of plant.

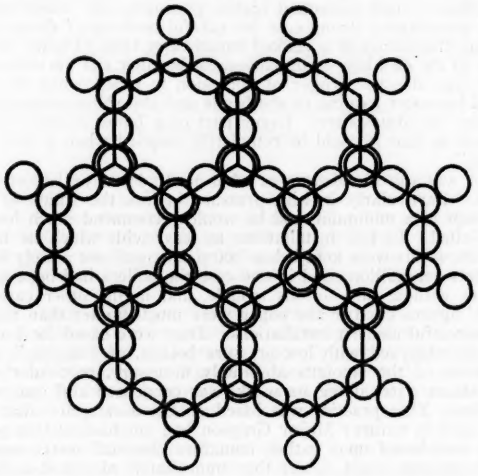

Figure 27.—Hexagonal Network of Molecules in Surface Film of Water (After Callendar).

Mr. Zoller had taken him to task for a remark which he had interposed, as a rather natural reference, to the similarity between two curves (Figures 2 and 3). Mr. Zoller, by replotting the curve of water-superheating in terms of excess vapor-pressure, instead of temperature, claimed to have arrived at some other inference, but did not state what it was. To probe the matter further inevitably led the discussion rather deeply into molecular physics, about which he himself did not pretend to have very extensive knowledge. Evaporation into the steam bubble spaces took place by the escape of molecules from the interfacial layer which, according to Professor Callendar's suggestion,* not improbably consisted of a hexagonal network of molecules (Figure 27) forming the "semi-permeable membrane required in osmotic theory, through which single vapor molecules could escape freely." It was in that monomolecular layer that practically the whole of the surface tension energy existed. Both surface tension, and the molecular energy (superheating) required by vapor molecules to penetrate that semi-permeable layer, would depend, he thought, upon the "diminution of the proportion of complex molecules with rise of temperature" in the surface layer. "The vanishing of the surface tension" (and the superheating of the water) "at the critical point would follow when the proportion remaining was insufficient to form a continuous network." He therefore contended that his method of plotting the superheating of the water in terms of its temperature was correct, and that on that basis one would expect some connection between the amount of water-superheating and the surface tension.

He purposely qualified nuclear evaporation, at a plane surface, by the word "theoretically." Mr. Zoller criticized the statement, but omitted that qualification and brought in the question of the conditions at the critical state, which only added to the hypothetical nature of the argument. Experimenters had confessed to the difficulty of observing the manner of the change of state at the critical temperature and pressure.†

Mr. Zoller also questioned the constancy of the rate of heat transmission through the bubble film, and quoted some figures from Jakob's paper, which he himself had not been able to trace. The explanation might, however, be provided by Figure 28, reproduced from Jakob's paper.‡ Jakob described

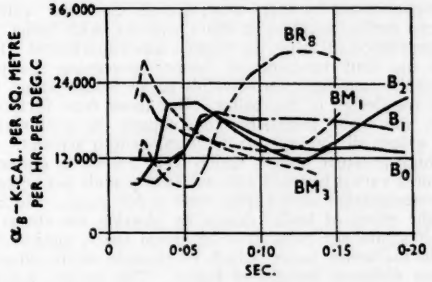

FIGURE 28.—COEFFICIENT OF HEAT TRANSFER FROM WATER TO STEAM BUBBLES (AFTER JAKOB).

Each curve, with its code letter, refers to the results for a particular bubble.

* Callendar, H. L., 1929, Proc. I. Mech. E., p. 819.
† Callendar, H. L., 1929, Proc. I. Mech. E., p. 814.
‡ Jakob, M. 1936, Mechanical Engineering, vol. 58, p. 656 (Figure 18); see also 1937 University of Illinois Engineering Experiment Station Reprint No. 10, p. 37 (Figure 18).

the graphs as relating to the rising of six bubbles in water. Jakob stated: "The fluctuations and deviations of the curves of α_B of Figure 28 are partly caused by the difficulties of evaluation. The mean values for the complete time of rising lie between 2600 and 4540 Btu. per square foot per hour per degree F., the average being $\alpha_B = 3200$ Btu. per square foot per hour per degree F." He himself did not expect that the extent to which the transmission rate might possibly vary with the pressure, for the reasons cited by Mr. Zoller, would invalidate the general deductions which had been made on the assumption that it would remain constant.

In a boiler, as Mr. Zoller rightly pointed out, some of the bubbles would jostle and touch one another, causing local hummocks of froth or emulsion, especially immediately above the submerged riser tube openings in a drum, but that would not occur evenly over every square centimeter of water surface; if every bubble threw up a droplet, and it was carried away by the steam, the proportion of moisture carried over would be enormous. Again, steam-and-water mixture impinging on baffles would not provide the same opportunity for the droplet action to take place. It was, in the last recourse, a matter of probability; any condition which increased the probability that a proportion of the steam bubbles would project droplets, would conduce to wetness of the steam.

Mr. Hewson's contribution was most interesting, and indicated what *could* be accomplished in the control of boiler water. Nevertheless, he himself knew of cases in which appreciable carry-over of solids had occurred even with concentrations of less than 300 p.p.m. in the boiler water, although all the usual mechanical precautions had been taken to ensure dry steam. The chemical constitution of the dissolved material evidently had some influence on the result.

Mr. G. A. J. Begg remarked that during the last few years he had been concerned with high-pressure boilers supplied with chemically treated water instead of the distilled water customary in central power station practice. The problem of carry-over had been very real and had caused considerable inconvenience in the operation of the turbines. He had attempted unsuccessfully to sift the American and German technical press with the object of finding out the causes of the carry-over; he had found that certain stations were inconvenienced by carry-over, though the boiler water was apparently of extreme purity, whereas in other stations, with boiler water with a very high concentration of solids, no trouble was experienced. He concluded, however, that the four fundamental factors governing the problem were steam pressure, the concentration of solids in the boiler water, the nature of the solids and the design of the boiler. But those four factors were interrelated, and it was quite impossible to compare the performances of different boilers unless all four factors were taken into account. He had with him a list of high-pressure boilers which were known to operate with satisfaction; pressures varied between 450 and 1850 pounds per square inch. The boiler water concentration varied from 6000 to 500 p.p.m. He had attempted to determine the effect of boiler design by plotting the steam release rate, in cubic feet of steam per cubic foot of steam space, against the total dissolved solids in the boiler water, which he thought might afford some comparison between different designs of boiler. The results, however, showed no consistency, whether for boilers of the orthodox type or for boilers of the forced circulation type. In certain cases boilers with steam scrubbers showed worse tendencies to carry-over than those without. In some instances the forced circulation boilers were much less prone to carry-over, presumably because of their more quiescent ebullition, and the more uniform separation of the steam from the water. But this did not always apply. He had further attempted to plot the kinetic energy of the rising steam against the specific volume of the steam by substitution in the factor ($V^2 \div$ specific volume of

steam), V being the velocity of the steam rising to the surface. He found that the numerical value of that fraction varied in practice from 0.01 to 0.06, which again gave no hope of a conclusion. He therefore welcomed the paper as a valuable contribution to the solution of a complex problem.

Dr. R. W. BAILEY said that the troubles caused by carry-over in boiler plants in this country were small in comparison with what occurred in America, where the process of steam washing had originated. He wondered whether the reason for the greater troubles encountered in America could be derived from Figure 22. In America boiler pressures had jumped from more moderate amounts to 1200 pounds per square inch, with 700 degrees F. steam temperature, whilst in this country the tendency in boiler development had been towards increasing the temperature and not so much towards raising the pressure.

Mr. R. CARSTAIRS said that he was associated with certain of the boiler plants with which Mr. Begg was concerned. He had found that the problem of carry-over of solids into the turbines appeared to be largely governed by the concentration of caustic alkali in the boiler water. The length of run of the turbines in one station was considerably longer than the length of run of similar turbines working under similar steam conditions in another station where the concentration of caustic alkali in the boiler water was some four times that of the first-mentioned.

Mr. R. J. ELDRED remarked that mechanical priming could be overcome by careful design, particularly when the aim was to produce correct circulation.

With regard to carry-over, which was primarily due to minute drops projected from the steam space, his firm had carried out various tests on commercial bled-steam evaporators. These were fitted with lamps and inspection glasses, which enabled the conditions at the surface to be observed. During the tests, visual observations were made of the approximate heights of the spray and splash. It had been noticed that there were two distinct forms of drops above the surface of the liquid: the large "fountain splash" like the drop of milk (Figure 16, Plate 1) about 2 millimeters in diameter, and in addition to much smaller drops like a fine spray. Most of the tests were carried out at 20 pounds per square inch absolute, and the results were shown in Figure 29. The curve had been plotted from formula (19), and there was good agreement with the splash observations. As a check, the curve had been replotted, using a form factor of 0.47, which was in line with

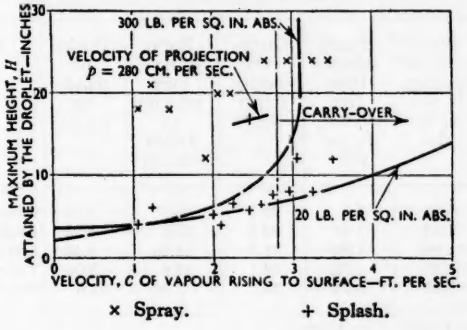

FIGURE 29.—CARRY-OVER IN AN EVAPORATOR.

modern aerodynamic information. To his surprise the new plotting made absolutely no difference. The author's initial velocity of projection was then doubled, and one point on the curve was indicated in Figure 29. That point lay very much nearer the zone of the spray observations. He suggested that the curves in Figure 29 might be of practical use for boilers if they were recalculated for an initial velocity of 300 centimeters per second. The height of the steam space in the evaporator concerned, which was of the vertical-coil type, was about 5 feet, yet if the steam velocity upwards exceeded about 2.8 feet per second, carry-over took place. According to the author (formula (18)), the carry-over velocity at 20 pounds per square inch absolute should be 13.5 feet per second. In an evaporator the steam dome height available was considerably greater than in a high-pressure boiler drum, which would appear to indicate the need for a much greater margin than a factor of safety of 4, on the carry-over velocity, calculated by the author's formula. The higher velocity of projection which apparently was responsible for the spray might be due to wavelets rather than to the mere bursting of bubbles. He did not think that the surface in an evaporator was any more disturbed than in a boiler; it was fairly uniform, broken up by wavelets, none of which exceeded about 2 inches in height.

Mr. E. V. WINSTANLEY remarked that he was chiefly interested in evaporators for make-up water for boilers. As evaporators were smaller and simpler than boilers, it was possible to use them to test the theories given in the paper.

For some time his firm had fitted observation windows with internal illumination on a number of installations and had observed the formation of foam and the way in which drops were thrown up. They had measured the purity of the vapor produced, and had correlated it with the concentration of the boiling solution.

The author made reference to the threshold values of concentrations at which soluble salts began to stabilize steam bubbles. Table 2 gave observations taken on an installation which substantiated the author's figures for sodium carbonate. Evaporation took place with the horizontal tubes "drowned" when boiling. Raw water was treated with lime-soda and calcium zeolite, so practically all the salt in solution was sodium carbonate. The free surface of the evaporator was 6 feet 4 inches by 16 feet 6 inches.

Table 2 showed that at concentrations above 3200 p.p.m., the formation of foam was commencing, and this appeared to support Foulk's figure of 3946

TABLE 2.—STEAM FORMATION IN AN EVAPORATOR.

Brine concentration, p.p.m.	Pressure of boiling, lb. per sq. in. abs.	Steam velocity from free surface, ft. per sec.	Carry-over velocity, ft. per sec.*	Ratio carry-over velocity to steam velocity	Purity of make-up p.p.m.	Remarks
3650	5.76	1.5	25.4	16.9	5.15	9 inches of foam
3200	7.40	1.63	22.8	14.0	4.55	4 inches of foam
2740	8.64	1.90	21.2	11.15	4.40	Little foam
2310	5.35	1.22	26.4	21.6	2.00	Trace of foam
2960	5.52	3.74	26.1	7.00	16.6	No foam†

* See formula (18), $d_o = 1$ mm.

†No foam, but heavy "spitting" priming occurred.

p.p.m. It was interesting to note that the last row of figures referred to conditions in which priming was taking place. Here the ratio of steam velocity to carry-over velocity (from formula (18)), was as low as 7.

Reference was made to a permissible rating based on a factor of safety of 4 for boiler work. For an evaporator, however, a considerably higher factor of safety was advisable if water of high purity was needed. It was, of course, appreciated that the tests referred to were carried out under commercial conditions, and that what the author called mechanical priming was always a possibility to be guarded against.

The author mentioned that Jakob had found that the rate of transmission from the water to the steam bubble tended to reach a steady value of 3000 Btu. per square feet per hour per degree F. This rate was of the same order as the apparent maximum rate of condensation of steam on a brass tube when "dropwise" formation was taking place; values of 2800 Btu. per square foot per hour per degree F. had been obtained at temperatures of the order of 120-180 degrees F. That rate of heat transmission could be accounted for by the thermal conductivity of a film of water 0.0017 inch thick, such as might form an envelope of water round the bubble of steam.

Mr. W. J. DICKIE observed that the author had described the theory of flow of water in a natural circulation boiler. The mechanical aspect of carry-over had not been sufficiently stressed. Today, manufacturers were adapting boilers which had been quite successful in the past for the generation of low-pressure steam, to enable them to carry higher pressures. Was that not a reason why carry-over was taking place to such a large extent? Too many devices had to be put into the boiler drum to remove the impurities which would otherwise be carried out with the steam.

It would be interesting to know how steam was formed under the influence of forced circulation. It was quite possible that only film boiling took place, but no tube failure occurred because the film, together with the water, was moving at a high rate. That was borne out by many years' experience with high-pressure boilers not provided with drums. No overheating or tube failure had occurred in spite of the fact that the tubes were mostly arranged horizontally in banks, falling from high to low levels.

Partridge and Hall (1939) had suggested that the circulation at pressures exceeding 1200 pounds per square inch was of paramount importance; failures of long tubes had been attributed to the fact that circulation had been impeded and had thus allowed steam blanketing to occur.

Carry-over was still more important in the boilers usually employed with high-pressure turbines, as the blade clearances for such turbines were sometimes smaller than those of equal output, but working at low pressures, and they were therefore more likely to become choked. He had had experience with boilers operating at 1400 pounds per square inch, supplied with feed water having a solid content of between 1 and 2 p.p.m. Even so, the turbines had become choked, and he had concluded that with turbines, carry-over if it were soluble, was not a serious trouble. He had found the solubility of the turbine blade deposit to be about 80 per cent, and it could easily be removed by washing with saturated steam. The insoluble portion was then carried away mechanically. Perhaps boiler manufacturers might consider washing out the superheater in much the same way as a turbine was washed out. It was formerly common practice to flood superheaters before they were started up, and thus any heavy carry-over which might have been deposited in the superheater would thus be dissolved by the washing-out water. No ill effect was caused. That gave support to his contention that superheaters should be designed so that they could be easily washed out.

The author's reference to the deposition of the semi-fluid at high temperatures and pressures had been borne out by evidence derived from a turbine operating at 1300 pounds per square inch; the semi-fluid was definitely sticky,

and in his opinion the other solids had become attached, and so built up the deposit on the first few wheels.

He had with him a sample of a tube from a high-pressure boiler operating at 1400 pounds per square inch with feed water having a total solid content of between 1 and 2 p.p.m. Deposit had occurred on the inside of the bend where the velocity would be lowest. The outside of the bend was practically clean.

Mr. A. RIDINGS remarked that the smaller size of the steam bubbles formed in high-pressure plant was the direct cause of the slower circulation known to exist in such units. That was indicated by Figure 11. From one aspect of the phenomena, it followed that the same factor of safety against overheating was obtainable with increased heat transfer in the riser tubes of high-pressure boilers, as could be obtained in medium-pressure units. Thus, apart from the possible formation of larger steam bubbles, with such advantages as the author claimed for consequent conditions of steam release at the water surface, an increased steam content (and hence reduction in density) of the bubbles would increase the rate of circulation.

Nevertheless, consideration must also be given to the possible effect on circulation of such factors as the greater adhesion of steam bubbles to the heating surfaces at high pressures, the varying value of such adhesion with differing tube bores, the distinct possibility of steam bubbles rising in a spiral path, and thus retarding circulation, and the restriction of circulation by the presence of "locked" steam bubbles or pockets which soon became highly superheated.

Having regard to all the circumstances, it would appear to be no safer to install long combustion chamber walls and front-bank steam generating tubes in high-pressure boilers than in units working at lower pressures and in this connection recent efforts to pack as much heating surface as possible into the radiant heat zone seemed to constitute a divergence from the best practice. Though closely spaced tubes offered mutual protection, very long tubes were subject to overheating, bulging, hogging, etc., and the actual amount of expansion (usually greater than the theoretical amount) was difficult to accommodate.

The data given in the paper emphasized the claim made by some boiler designers that generating tubes should be inclined as steeply as possible if the most rapid and free circulation was to be obtained, and overheating of the tubes avoided. The comparatively sluggish circulation in slightly inclined tubes must lead to surface boiling and high superheating, with distinct risk of overheating and hogging of the tubes, if not actual failure. He agreed that the screen tubes, referred to specifically in the paper as having failed, were subjected to more severe conditions than ordinary steam generating tubes, but the latter were nevertheless needed for onerous conditions. Impingement of hot gases on the top of such tubes, when in a high-temperature zone, should be avoided.

Though the volume of steam passing through the water surface of high-pressure plant was much less than in medium-pressure plants, and the surface area was (allowing for the size of the unit) not so severely taxed in regard to steam release, ebullition, carry-over, and the like, the size of the front steam and water drum of all three drum boilers (considered over the full range of pressures) might well be increased with great advantage to the quality of the steam. Present-day practice seemed definitely to err on that important point, the size of the drum being restricted to the requirements of the tube layout, rather than with reference to the steam release in the vessel, in which as much as 90 per cent of the total steam release occurred.

Mr. R. F. DAVIS said in reply that he agreed with Mr. Begg that it was difficult to correlate the occurrence of steam contamination with the four variable factors, namely, the steam pressure, the concentration of solids in the boiler water, the nature of the solids, and the design of the boiler. In the paper he had attempted to study the influence of each factor separately

although, as Mr. Begg had pointed out, they could not be so easily differentiated in practice. There was a considerable field for scientific and technical research on these problems.

Dr. Bailey asked why so little trouble from steam contamination was encountered in this country, in comparison with the U. S. A. The answer was provided by a table, published during 1937,* of the boiler units installed in the U. S. A. since 1st January 1936. During the period under review 61 boiler units, having a total capacity of nearly 25 million pounds of steam per hour were installed in central stations alone, at operating pressures exceeding 800 pounds per square inch; of those, the pressures of 33 were above 1000 pounds per square inch. Since then many more highly rated high-pressure boiler units had gone into operation in the States; thus the sum total of American experience in this field was considerably greater, and collected over a longer period, than our own. Added to this was the frankness with which American engineers published their difficulties.

The definite influence of the presence of caustic alkali in the boiler water on the carry-over of solids into the superheater and turbine had been exemplified by the experience of Mr. Carstairs. That the proportion of caustic, rather than the total concentration, also had a definite influence, had only recently again been brought to his notice. A boiler steaming with less than 300 p.p.m. total dissolved solids in the boiler water and with only about 0.5 per cent carry-over of moisture in the saturated steam had suffered, after a comparatively short period of service, from choking by magnetic iron oxide in the superheater tubes. In this case, more than half of the dissolved solids in the boiler water was sodium hydroxide. Overheating and rapid internal corrosion of the superheater tubes had occurred.

The interesting figures contributed by Mr. Eldred and Mr. Winstanley on the performance of evaporators were an important extension of the principles enunciated in the paper, to much lower operating pressures than he himself had in mind when writing the paper. There was no reason why the same laws should not be applicable within that range, but obviously some of the constants which he had used in the formulae, with a view to their application to high pressures, would need modifying for low pressures. The factor of safety of 4 which he had used to obtain the recommended permissible rating (Figure 23) was based on experience of all the boilers of whose performance he had records. The curves of Figure 22 must be taken as illustrating the qualitative aspect of the influence of pressure on the carry-over of drops, rather than as representing absolute velocities, because absolute values depended on constants which could only be satisfactorily determined by research. Such research required very elaborate apparatus to take photographs at a rate of a thousand per second or more inside a high-pressure vessel.

Mr. Dickie's opening remarks seemed to imply that nuclear boiling did not exist with forced circulation; if this was Mr. Dickie's meaning, he could not agree with him. Forced circulation could be made more rapid than natural circulation, and the quicker circulation would sweep the bubbles from the heating surface while they were still very small. Mr. Dickie's experience of turbine deposits by carry-over from "once-through" type forced circulation boilers, showed that even that type of boiler was not immune from steam contamination, notwithstanding feed water of the extreme purity of 1 or 2 p.p.m. of solids. However, Mr. Dickie did not seem very concerned by the deposits in his turbines, as they were so easily washed out; unfortunately, for the boiler manufacturer, not all power plant engineers were in such a happy frame of mind about such deposits.

The deposits formed in superheaters were also mostly soluble and could usually be washed out, which was the method of removal frequently employed. The difficulty was that the first indication the deposits gave of their

* Power, 1937, vol. 81, p. 589.

presence was the failure of several tubes; whereas in a turbine a falling-off of efficiency indicated that something was wrong; the need for washing out was thus known beforehand, before any damage could result.

The smaller size of the steam bubbles at higher pressures was not the only cause of slower circulation, as Mr. Ridings remarks might seem to suggest. The more sluggish rate of separation of the smaller bubbles from the water increased the chances of their entrainment in the downcomers. The thermo-syphonic head producing circulation was thus reduced, unless precautions were taken to increase the chance of separation of the bubbles by arranging for the impingement of the steam-and-water mixture on to baffles. The decrease in the density differential of the steam and water (Figure 1) was perhaps a more potent factor influencing circulation, but even this was not so serious in its effects as might be imagined.* That was borne out by practical experience, for example, up to the present time, all high-pressure water-tube boilers in the U. S. A. had been of the natural circulation type.

It was not clear why Mr. Ridings thought that the steam bubbles would have greater adhesion to the heating surface at high pressures; the decrease in both the surface tension and the viscosity of the water at increased saturation temperatures would seem to promote the reverse effect. Perhaps Mr. Ridings had in mind the reduced rate of circulation.

Evidently Mr. Ridings did not favor the modern type of boiler with considerable radiant heating surface consisting of long tubes; but many hundreds of such boilers had been built, and had been in satisfactory operation for several years. The reasons for this trend in the development of water-tube boilers had been discussed elsewhere (Davis and Timmins).* Larger drums for high-pressure boilers would, no doubt, give improved steam quality, but the cost and the practical difficulties of manufacturing drums of the necessary thickness to withstand the pressure made them prohibitive. A better plan was to use two small drums, or, as was more often done, to incorporate an additional smaller drum above the steam-and-water drum to act as a steam receiver and to provide an additional stage for moisture separation.

Summarizing the discussion, there had been but little criticism of the subject matter of the paper. Some valuable contributions to it had been made, but there had been some questioning of the aptness of modern boilers to meet the conditions of service now often imposed upon them. He hoped, however, that in future, when steam contamination occurred, the condition of the boiler water would not be ignored to the disparagement of the design of the boiler. Designers and power plant engineers were at the present day aiming at a steam purity in high-pressure boilers of the order of less than 0.25 per cent moisture content and 0.25 p.p.m. of solids, and to accomplish this demanded the close co-operation of both parties.

COMPARISON OF MARINE DIESEL PROPULSION DRIVES.

Mr. Wesley Moore, Manager, General Diesel Sales, Fairbanks, Morse and Co., is author of this article reprinted in entirety from the August, 1941, issue of Motorship and Diesel Boating. The relatively high cost and weight of Diesel electric drive is notable.

Engine Speeds.

For the sake of brevity, this considers only the more popular classes of vessels requiring more than 300 shaft horsepower for propulsion. Although each particular marine application must be studied separately, certain gen-

* Davis, R. F., and Timmins, A. L., 1933, Proc. I. Mech. E., vol. 125, p. 732.
* See also Davis, R. F., 1937, Proc. I. Mech. E., vol. 137, p.44.

eralities exist in the relationship between shaft horsepower, propeller revolutions and vessel speed. For example, free running vessels may effectively utilize much higher propeller revolutions than can tugs, operating at slower towing speeds. What therefore becomes, in connection with direct drive, a high engine speed applied to a tug, might likely be called a very moderate speed when applied to a yacht. The accompanying curves in Figure 1, have been plotted to show a limiting relationship between shaft horsepower and propeller speeds for several typical installations. These curves must not be taken too literally, as they are affected by each individual design. For a given type and speed of vessel, however, they do indicate quite clearly the need for lower revolutions as horsepower is increased. They also show the possibility of utilizing higher revolutions with a given horsepower in vessels designed for higher speeds.

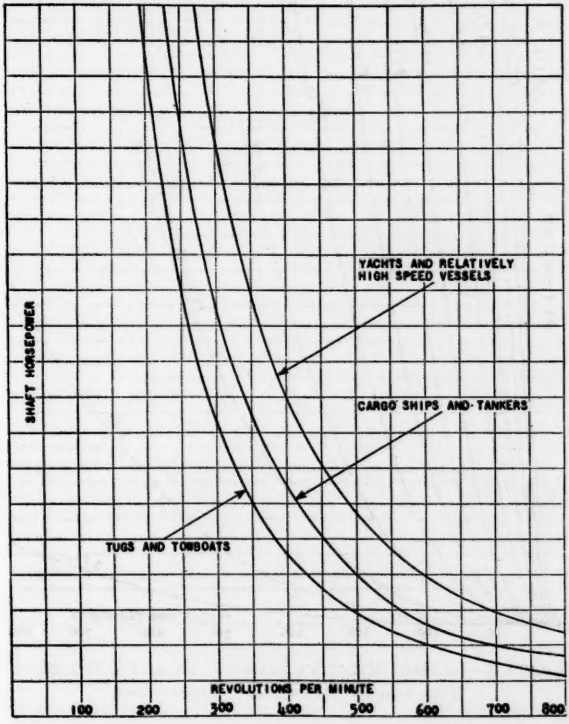

FIGURE 1.—LIMITING RELATION BETWEEN SHAFT HORSEPOWER AND PROPELLER SPEED FOR TYPICAL INSTALLATIONS.

PROPELLER DIAMETERS.

The advantages of low propeller revolutions applied to tugs are well recognized but frequently where either a geared or electric Diesel drive

is selected and one has the free choice of propeller revolutions, the limiting factor may be the maximum diameter propeller that can be accommodated within the propeller aperture. Although propeller diameter, propeller revolutions, and shaft horsepower, vary with a given design, the curves in Figure 2 show the general relationship indicating necessity for quite large diameter propellers when revolutions are carried too low. Also, unreasonably small diameter propellers when revolutions are carried too high. Because of limited draft, river towboats using tunnel screws, for example,

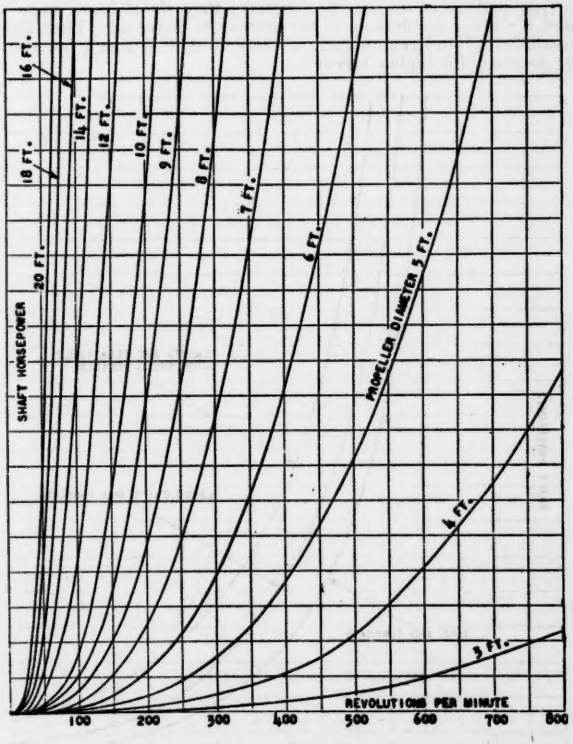

FIGURE 2.—GENERAL RELATION BETWEEN PROPELLER DIAMETER, REVOLUTIONS AND SHAFT HORSEPOWER.

rarely can accommodate propeller diameters in excess of eight feet. Accordingly, for any powers in excess of 1000 horsepower it has been customary to use twin screws. Recently constructed harbor and East Coast tugs which have employed electric drive with single screws up to 1000 horsepower are limited in most cases to maximum propeller diameters of about nine feet because of partial operations on the New York State Barge Canal. It is reasonable to presume that if larger diameter propellers could

DIESEL ELECTRIC DRIVES

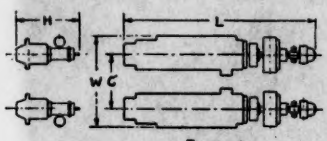

FIG. No. 4

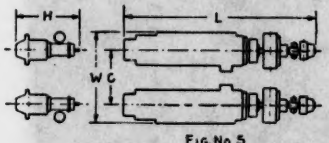

FIG. No 5

NOTE: COMPARATIVE COSTS, WEIGHTS, AND SPACE, IN EACH CASE,

SHAFT HORSEPOWER	PROPELLER SPEED-R.P.M.	FIGURE No.	L-O.A. Length	W-O.A. Width	H-O.A. Height	C-c-c units	2-ea Length motor	W-ea Width motor	COMPARISON OF COST	WEIGHT PER HP @ Pounds	SPACE PER HP @ Cubic Feet	DESCRIPTION DIESEL PLANT
360	300	4	16'-0	10'-4	5'-11½	6'-4	6'-10	5'-2	UNITY	167.5	3.05	TWO 225 HP DIESELS D.C. GENERATORS
550	240	4	16'-6	10'-4	5'-11½	6'-4	7'-8½	5'-8	UNITY	163.0	3.18	THREE 225 HP DIESELS D.C. GENERATORS
800	200	4	23'-1	13'-6	9'-2	8'-6	8'-7½	6'-10	UNITY	206.0	4.26	TWO 500 HP DIESELS D.C. GENERATORS
1000	200	4	27'-6	13'-6	9'-2	8'-6	9'-0½	8'-2	UNITY	177.2	3.89	TWO 600 HP DIESELS D.C. GENERATORS
1400	150	4	32'-2	17'-0	11'-9	10'-6	10'-11½	10'-6	UNITY	228.8	5.18	TWO 875 HP DIESELS D.C. GENERATORS
1750	150	4	35'-1	17'-0	11'-9	10'-6	11'-1½	12'-3	UNITY	213.0	4.60	TWO 1050 HP DIESELS D.C. GENERATORS
2000	175	5	35'-8	17'-0	11'-5	10'-6	8'-11	8'-2	UNITY	210.1	3.97	TWO 1225 HP DIESELS D.C. GENERATORS
3000	125	4	39'-6	17'-6	10'-10	10'-6	12'-7	13'-1	UNITY	196.0	3.09	TWO 1750 HP DIESELS D.C. GENERATORS
3500	90	4	34'-2	35'-0	11'-9	10'-0	14'-1	11'-5	UNITY	213.0	4.63	TWO 2100 HP DIESELS D.C. GENERATORS

NOTES:

1. WEIGHTS AND COST COMPARISON COVER ENGINES WITH MA ENGINE EQUIPMENT AND ALSO INCLUDE GENERATORS, EXCITERS PULSION MOTORS AND VARIABLE VOLTAGE CONTROL EQUIPMENT.
2. SPACE REQUIREMENTS INCLUDE SPACE REQUIRED FOR DIES GENERATING UNITS AND PROPULSION MOTORS, BUT DONOT I SPACE REQUIRED FOR CONTROL EQUIPMENT
3. FIGURE No. 4 COVERS TWIN ENGINE SINGLE SCREW ARRANGEME
4. FIGURE No 5 COVERS TWIN ENGINE TWIN SCREW ARRANGEME
5. COVERS OVERALL WIDTH OF THREE GENERATING UNITS.
6. COVERS OVERALL WIDTH OF FOUR GENERATING UNITS.

FIGURE 3.—COMPARISON OF THREE TYPES OF D

GEARED DRIVES

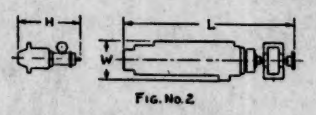

FIG. No.2

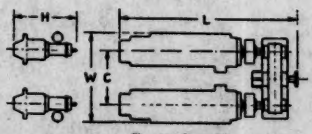

FIG. No.3

DIRECT DRIVES

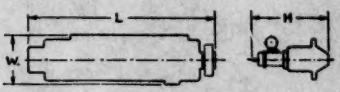

FIG. No.1

, IN EACH CASE, CORRESPOND TO "AVAILABLE" HORSEPOWER AT PROPELLER SPEEDS, ALLOWING FOR ALL LOSSES.

SPACE PER HP (D) Cubic Feet	DESCRIPTION DIESEL PLANT	SHAFT HORSEPOWER	PROPELLER SPEED - R.P.M	FIGURE No	L - O.A. Length	W - O.A. Width	H - O.A. Height	C - c.l.c. cads	COMPARISON OF COST	WEIGHT PER HP (D) Pounds	SPACE PER HP (D) Cubic Feet	DESCRIPTION DIESEL PLANT	SHAFT HORSEPOWER	PROPELLER SPEED - RPM	FIGURE No	L - O.A. Length	W - O.A. Width	H - O.A. Height	COMPARISON OF COST	WEIGHT PER HP (D) Pounds	SPACE PER HP (D) Cubic Feet	DESCRIPTION - DIESEL PLANT
3.05	TWO 225 HP-720 R.P.M DIESELS D.C. TO 150 K.W GENERATORS	315	300	2	21'-2"	4'-10"	6'-5"		.504	130.0	2.04	ONE 320 HP 400 R.P.M DIESEL	375	300	1	17'-6"	5'-6"	5'-2"	.417	142.5	2.42	ONE 375 HP 300 R.P.M DIESEL
3.18	THREE 225 HP-720 R.P.M DIESELS D.C. TO 150 K.W GENERATORS	592	240	3	18'-4"	11'-6"	6'-6"	7'-7"	.511	98.2	2.05	TWO 300 HP 720 R.P.M DIESELS	644	240	1	24'-6"	6'-4"	10'-6"	.440	133.3	2.48	ONE 644 HP 240 R.P.M DIESEL
4.26	TWO 300 HP-400 R.P.M DIESELS D.C. TO 200 K.W GENERATORS	690	200	2	27'-0"	5'-8"	9'-2"		.531	111.3	2.04	ONE 700 HP 400 R.P.M DIESEL	700	200	1	27'-0"	7'-6"	11'-9"	.646	164.0	3.40	ONE 700 H.P. 200 R.P.M DIESEL
3.80	TWO 600 HP-400 R.P.M DIESELS D.C. TO 420 K.W GENERATORS	970	200	3	24'-6"	13'-0"	9'-2"	7'-7"	.671	152.0	3.04	TWO 500 HP 400 R.P.M DIESELS	1000	200	1	28'-6"	7'-9"	11'-9"	.613	140.4	2.60	ONE 1000 HP 200 R.P.M. DIESEL
5.18	TWO 875 HP-300 R.P.M DIESELS D.C. TO 600 K.W GENERATORS	1360	180	3	28'-6"	13'-0"	9'-2"	7'-7"	.594	131.5	2.53	TWO 700 HP 400 R.P.M. DIESELS	1500	150	1	28'-6"	16'-6"	17'-6"	.735	181.0	2.97	ONE 1500 HP 150 R.P.M DIESEL
4.60	TWO 1050 HP-300 R.P.M DIESELS D.C. TO 720 K.W GENERATORS	1665	150	3	31'-6"	17'-6"	11'-5"	10'-6"	.672	154.0	3.82	TWO 875 HP 300 R.P.M DIESELS	1750	150	1	31'-6"	18'-6"	17'-6"	.779	185.0	2.83	ONE 1750 HP 150 R.P.M DIESEL
3.97	TWO 1225 HP-300 R.P.M DIESELS D.C. TO 840 K.W GENERATORS	2000	125	3	33'-6"	17'-6"	11'-5"	10'-6"	.654	134.8	3.40	TWO 1050 HP 300 R.P.M DIESELS										
3.09	TWO 1750 HP-300 R.P.M DIESELS D.C. TO 1200 K.W GENERATORS	2850	125	3	36'-6"	18'-7"	11'-5"	11'-6"	.684	133.6	2.90	TWO 1500 HP 300 R.P.M DIESELS										
4.63	FOUR 1050 HP-300 R.P.M DIESELS D.C. TO 700 K.W GENERATORS	3325	90	3	39'-6"	18'-6"	10'-5"	11'-6"	.539	153.5	2.45	TWO 1750 HP 300 R.P.M DIESELS										
		3325	90		31'-6"	36'-6"	11'-5"	10'-6"		160.0	4.24	FOUR 875 HP 300 R.P.M DIESELS										

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NOTES:

1. WEIGHTS. SPACE AND COST COMPARISON COVER ENGINES WITH MAIN ENGINE EQUIPMENT AND INCLUDE REDUCTION GEAR AND COUPLINGS REQUIRED BETWEEN ENGINE AND GEAR.
2. FIGURE NO 2 COVERS SINGLE ENGINE SINGLE SCREW ARRANGEMENT AND PROVIDES FOR FLEXIBLE COUPLING BETWEEN ENGINE AND REDUCTION GEAR.
3. FIGURE NO 3 COVERS TWIN ENGINE SINGLE SCREW ARRANGEMENT AND INCLUDES ELECTRIC SLIP COUPLINGS BETWEEN ENGINE AND REDUCTION GEAR FOR SIZES FROM 970 TO 3325 HP INCLUSIVE.
4. COVERS ARRANGEMENT WITH FLEXIBLE TYPE COUPLINGS BETWEEN ENGINES AND GEAR.
5. COVERS OVERALL WIDTH OF FOUR ENGINES AND TWO GEARS.

FIGURE 3 —

A comparison of Diesel-propulsion drives.

TYPES OF DRIVE; DIRECT, GEARED AND DIESEL-ELECTRIC, SHOWING RELATIVE COSTS, WEIGHTS AND SPACE.

be accommodated such as in sea-going tugs, there would result a wider acceptance of indirect drive for coastal tugs. Maximum draft and propeller diameter limitations however, frequently dictate a propeller speed sufficiently close to rated speed of the required engine to eliminate from consideration the added complications of an indirect drive.

Space and Weight.

No comparison of direct and indirect Diesel propulsion drives could be complete without reference to space and weight factors and, oddly enough, on many shallow draft vessels, requiring powers up to 2000 horsepower, space and weight limitations of propulsion machinery are quite liberal. On the Mississippi, Ohio and Illinois Rivers, for example, the towboat hulls must be quite large to provide for crews quarters and the necessary draft. Since these towboats do not carry cargo there is ample room in the hull for propulsion and auxiliary machinery and it is rarely necessary to crowd the machinery as with cargo-carrying boats. Perhaps this, coupled with the necessity for limited propeller diameters suggesting propeller speeds in most cases quite close to normal engine revolutions, accounts for the present lack of interest on the part of river towboat operators in indirect Diesel drives. Several very interesting indirect drives have been applied, however, in recent years to cargo-carrying vessels for use on the Lakes, and, for many applications, the use of either geared or electric drive permits of a much greater earning capacity for the owners. The table in Figure 3 gives space weight and price comparisons for three forms of drives in various powers at comparable propeller revolutions. While present-day prices covering electric drives show little tendency towards change, gear prices are continually being reduced as development expense is written off. Accordingly, this should be borne in mind in connection with any specific proposition.

Flexibility of Indirect Drives.

The use of indirect drive has certain advantages quite aside from those obtained through the use of relatively lower propeller speeds. In the case of the gear application, for example, the power of two or more engines may be combined to serve a single propeller and by providing a means of disconnecting each engine from its gear pinion, the flexibility of multiple units is made available. Where desirable, the propeller shaft may be offset somewhat from the plane of the engine crankshaft in any desired direction. In the case of the electric drive, the generating units need not be placed in any fixed relation to the propeller shafting and frequently the electric propelling motor may be placed to advantage on a flat by itself quite further aft with a much shorter shafting than would be possible with equal trim using either gear or straight drive application. In general, either straight drive or geared applications provide close governing only down to about 30 per cent of normal engine revolutions. Direct reversals are usually substituted for lower propeller revolutions when maneuvering in crowded waters. Electric drives employing direct current however, permits splendid speed control of the propelling motor for all speeds by remote controls placed at several convenient points on the vessel.

In spite of the many Diesel electric advantages, new construction within the past few years involving tankers and cargo carriers for operation on the Great Lakes has favored mechanical drives. For 300-foot tankers and cargo carriers operating coastwise and on the Lakes the choice has been about equally divided between direct drive and geared drive. Twin screws have been employed for vessels of this size and total propulsion power of about 1500 horsepower has been popular. Draft limitations have in most

cases limited propeller diameters to about seven feet and twin propellers supplied with about 750 horsepower each, have proved efficient at 300 revolutions per minute. Figure 4 shows relative dimensions of a straight drive 750-horsepower, 300-RPM. propulsion engine and 720-horsepower, 600-RPM. engine equipped with 2:1 gear reduction. Since twin engines of either type of drive would provide about equal power at the same propeller revolutions, the choice between the two would have to be made from other considerations. As is shown in Figure 4, the geared unit enjoys the advantage of lower weight and less space. The straight drive engine being of larger bore and stroke and slow speed could utilize a wider range of fuels. Fuel consumptions, however, would be about comparable and any saving in fuel operating cost would result from difference in price of fuels used. Maintenance expense could be expected to be higher for the geared combination and at least part of the advantage gained in greater carrying capacity must be sacrificed to idle ship time occasioned by increased repairs.

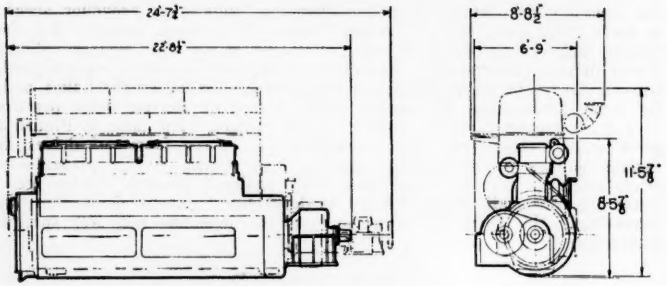


FIGURE 4.—RELATIVE DIMENSIONS OF DIRECT CONNECTED 750-HP. ENGINE AND A 720-HP. ENGINE CONNECTED THROUGH GEARS.

Most new coastal fishing boat construction has so far favored direct drive. Trawlers of about 150 feet in length operating from the Boston fish pier are mostly powered with about 600 horsepower on a single screw to give vessel speed of about 10 knots. Propeller speeds of between 200 and 300 RPM. have been popular. It is entirely possible that in the future new trawler construction will favor geared applications. Owing to the nature of the service, however, requiring engines of considerable reliability, with a minimum of time available for repairs, it is unlikely the owners will make too great a compromise in the direction of higher speed engines to gain added carrying capacity. While it is physically possible, for example, to use a multiplicity of high speed engines with a combined gearing to serve a single propeller it is unlikely that such an arrangement would be practical or prove as popular as one or two moderate speed machines to serve a single or multipinion gearing.

For 500-foot tankers operating coastwise between ports in the Gulf and East Coast, recent construction has favored direct drive engines. Such ships employ about 6000 horsepower and carry a 19-foot single propeller at about 90 RPM. These direct drive engines are of a similar type to that employed in many Maritime Commission motor vessels. Although much headroom is required by a direct drive engine of this capacity, the simplicity of the machine has a strong appeal from a practical viewpoint. Because of the easy tolerances possible with an engine of such low speed, prior assembly

for test purposes has been found unnecessary, the engine merely being assembled in the ship and given its initial start at the time of dock and sea trials.

Although electric drive has as yet not been employed for powers up to 10,000 horsepower in the Maritime Commission program, indirect gear drives have been used involving either two or four engines to serve a single propeller through combined gearing (see Figure 5). Some of the ships employing gear drive are using electric couplings and some hydraulic couplings. Both the hydraulic and electric couplings have splendid characteristics a few of which are worthy of mention. Each coupling has a certain amount of damping effect between engine and propeller shafting. Each coupling may be declutched. In the case of the electric coupling this may be done instantaneously and, in the case of the hydraulic coupling, in from 10 to 15 seconds. Each type of coupling is reasonably efficient, the loss with either being less than 3 per cent.

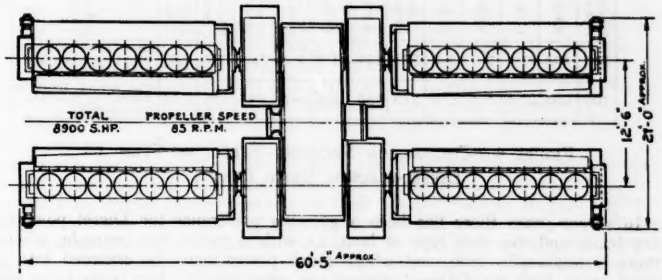
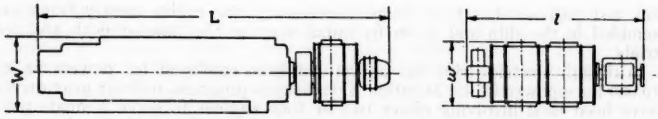


FIGURE 5—GENERAL ARRANGEMENT OF FOUR 2225 HORSEPOWER ENGINES DRIVING A SINGLE SHAFT THROUGH REDUCTION GEARS FOR C-3 SHIPS. TOTAL WEIGHT 1,005,000 POUNDS.

APPLICATIONS FOR DIESEL-ELECTRIC DRIVES.

One of the most popular applications for marine Diesel engines has been the driving of ferry boats. Although gear drive Diesels have not been popular, there have been some very outstanding installations involving Diesel electric drives. Until recently, most of these Diesel electric applications have involved moderate speed engines. The Diesel electric drive has been favored where the ferries must operate in crowded channels. Where the runs are long, however, straight drive has usually been favored. Space and weight of the generating units themselves has not been a major factor and since the straight drive Diesels have, in the majority of cases, employed double end drive, such as is used with recent Diesel electric motor-driven ferries, the choice between direct and Diesel electric drive lies largely between first cost and maneuverability, with the electric drive of course favoring the latter and direct drive favoring the former. In Figure 6 a comparison is shown between the first cost of two sets of Diesel electric machinery.

In each case, a 575 horsepower, 180-RPM. propelling motor is used with appropriate control. It will be noted that the total machinery is but 5 per cent less costly when this propelling motor is served with a 600-RPM. generator than when a 400-RPM. generator is used. Such a comparison suggests careful consideration for slower turning engines.



COMPARISON OF COST, WEIGHT AND SPACE OF DIESEL ELECTRIC MACHINERY CONSISTING OF 575 HP. PROPULSION MOTOR AND DIESEL GENERATING UNIT AT 400 R.P.M. CONTRASTED TO 600 R.P.M.

GENERATING UNIT					PROPULSION MOTOR				COMPARISON COST-WEIGHT-SPACE			
Diesel Engine Horsepower	R.P.M.	L-O.A. Length	W-O.A. Width	H-O.A. Height	Shaft Horsepower	R.P.M.	l-O.A. Length	w-O.A. Width	Comparison of Cost.	Weight per S.H.P. Pounds.	Space per S.H.P. Cubic Feet.	Description Diesel Plant.
700	400	27'-1"	5'-9"	9'-2"	575	180	13'-9"	4'-8"	Unity	197	2.87	One 700 H.P.-400 R.P.M. Diesel D.C. to 470 K.W. Generator
700	600	22'-5"	5'-0"	8'-6"	575	180	13'-9"	4'-8"	.948	159	2.05	One 700 H.P.-600 R.P.M. Diesel D.C. to 470 K.W. Generator.

FIGURE 6.—COMPARISON SHOWING EFFECT OF SPEED ON DIESEL-ELECTRIC DRIVE PLANT.

In recent years there has been a growing preference for Diesel powered fire boats and, for this type of boat, as with a yacht, for example, where there is much idle time, and where much power must be crowded into a small space, high speed Diesel engines are quite logical. For boats from 100 to 150 feet in length speeds of from 14 to 16 miles per hour are popular. Since the function of the boat is two-fold with the first purpose of getting to the fire in the least time and, when there, making the greatest amount of power available for the fire pumps, it has been customary to utilize the major portion of the installed power to drive the fire pumps. This can be done with clutches and multiple units by having the propulsion engines clutch connected at the forward ends to serve fire pumps, but a much more elastic arrangement is possible with Diesel electric machinery permitting one or more motor-driven fire pumps to be served by one or more Diesel-generator units supplying at the same time enough electrical power to the propelling motor to give headway.

Tuna boats operating from the southern coast of California present an interesting problem. These vessels must make long trips and, to preserve their fish when operating in warmer climates, must have considerable refrigeration machinery requiring a relatively large amount of electric auxiliary power. It is but a step further to increase the size of the generation equipment to serve an electric propelling motor as well as the auxiliary load.

CONCLUSIONS.

From the foregoing it will be appreciated that all three forms of drive have their advantages and the limitations of either when applied to a certain application may rule against its use. It may be somewhat of a temptation in considering indirect drives, either geared or electric, to figure on the use of light duty engines as the source of power. This is largely due to weight, space, or apparent first cost advantages. When it is realized, however,

that for a certain power, the cost of either the reduction gear or the electric machinery is very little affected by the speed of the main engine, what might otherwise be a rather substantial percentage difference between high and moderate speed engines, becomes but a nominal percentage difference on the complete machinery and, percentage-wise, applied to the complete vessel, of very minor proportions.

Much advantage can be shown in certain applications through improved propulsion results with indirect drive. These advantages may be had with either high or moderate speed engines. If an attempt is made, for example, to extend the power by use of higher engine revolutions the advantage, when applied to commercial vessels, may be more apparent than actual. No one can say what is a high speed, a moderate speed, or a low speed, without knowledge of the specific application.

That engines have been designed and produced primarily for certain specific applications ashore, does not necessarily qualify such engines for continuous heavy duty marine service. The many different designs of commercial Diesel engines which are available give striking evidence of the individuality of each builder. Some engines are two-stroke while others are four-stroke. Some are single and others are double acting. Piston travels, mean effective pressures, bore-stroke ratios, and other features of design will vary according to the individual builder's preference. Irrespective of the design, however, and with all other factors unchanged, the higher the rotative speed, the noisier the engine becomes and the more sensitive it is to the kind of fuel it can successfully burn.

Although there are very definite engine speed limitations tied in with the propeller requirements for direct drive, there has been a tendency on the part of some engine builders to rev up their smaller engines to give desired sizes for towboat service. While horsepower gained in this way, aside from any question of expected life, may be said to be test block power and transmittable onto the name plate, it is not necessarily usable or available power at the propeller when considering actual towing speeds.

For many towing operations a four-knot towing speed is the maximum that is practical and, for any given towing speed, there is a definite maximum propeller speed above which serious cavitation commences.

While reving up an engine may create the impression of a bargain in the way of low price per horsepower, the higher the propeller speed for a given towing speed, the smaller must be the propeller diameter, pitch, and resulting propeller thrust. This may be noted in the table.

Note that a towing propeller designed to absorb 500 BHP. at 300 RPM. will deliver 26 per cent more thrust than one designed for 400 RPM.

The speed of a towboat through the water directly affects the design of the propeller. If propeller is designed for six miles per hour and the weight of the tow is figured for seven miles per hour the propelling engine will reach the designed RPM. without using its full BHP.

This condition is satisfactory as most operators will not care to load the propulsion engine to its full capacity, but where the propeller designed for seven miles and the size of the tow limits top towing speed to six miles, the propelling engine would be over-loaded at full engine revolutions. This latter condition would of course prove harmful to any engine and suggests the danger of over-estimating a maximum towing speed when designing a propeller. This is mentioned since in conjunction with certain towing operations some may attempt to justify over-reved straight drive engines for towing service in expectation of higher towing speeds.

Summing up, it has been attempted to bring out the advantages of each type of the three drives, direct, geared and electric. Each of these drives, within its limitations, has a definite place in the future of the marine Diesel industry. As brought out, none of these drives can be any better, however,

than the main engine. It is only natural, and to be hoped for, that there will be a wider acceptance in the future of indirect drives. Such acceptance should result from the merit of the drive itself, however, and not at the expense of excessive Diesel ratings. Otherwise, high Diesel upkeep expense would soon label Diesels impractical for merchant vessels.

Engine b.hp.	500	500	500	500
Towing speed	4 knots	4 knots	4 knots	4 knots
RPM	100	200	300	400*
Propeller dia., in.	147	97	75	64
Propeller pitch, in.	86	54	41	32
Theoretical thrust hp.	179.5	144	126	100
Cavitation at	173	245	305	400

* Since cavitation commences at 400 RPM, greater propeller blade area has been assumed for this last condition. This required increased area causes the blades to overlap somewhat.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

ULTRASONICS.—Several applications of supersonic waves are mentioned in this item reprinted from The Metallurgist, August, 1941, supplement to The Engineer, August 29, 1941.

Much attention has of late years been devoted to the possible applications of ultrasonic or supersonic waves. These waves are mechanical vibrations having a frequency between about 20 kHz and some undefined upper limit of many thousand kilocycles. The outstanding problems associated with their successful application are the construction of suitable generators for the waves and the development of a sensitive detector. A short article, prompted by the announcement that the first American ultrasonic generators are now commercially available, is contributed by C. M. Cosman[1] to the Iron Age. Beyond an illustration of an 80-watt equipment made by the Televiso Products, Inc., 2400, North Sheffield Boulevard, Chicago, for producing waves of 400 kHz frequency, no details are given and no mention is made of a commercially produced detector. There are, of course, many possible applications of ultrasonics for which a detector is not required, but if they are used in place of X-rays or gamma rays from radium a sensitive detector is essential.

In an article[2] in The Metallurgist in 1938 references were given to applications of supersonic waves which included their use for under-water signalling, the detection of icebergs and location of submarines. Other effects referred to were the liberation of gases from liquids, with possible application to the degassing of molten metals, the avoidance of superheating, the crystallization of supersaturated solutions and the refinement of grain size in solidification, the diminution in the passivity of iron, the speeding up of the nitriding of steel and of the aging of duralumin, the emulsification of liquids, the dispersion of lead in molten metals, the preparation of colloidal solutions and, on the other hand, the agglomeration of particles in fogs and in industrial dust. The application of supersonic waves in non-destructive testing was also mentioned.

Since that time several valuable general accounts of the use of ultrasonics have become available,[3] including descriptions of the means by which they may be applied to the non-destructive testing of metals as an adjunct to or substitute for radiographic examination. The outstanding contributions to this aspect of the subject have been made by Sokoloff and those associated

with him. In Sokoloff's original form of detector the emergent beam was observed by characteristic ripples on the surface of a liquid in contact with the specimen. The vibrating surface of the liquid acts as a diffraction grating in reflecting light and a flaw in the material produces a cloudy patch on a screen or photographic plate receiving a beam of light reflected from the liquid surface. Another form of detector has been described by R. Pohlman.[4] In this, fine flakes of aluminum (about 1.5μ thick and 20μ diameter) are suspended in xylene contained in a flat cell, one side being of copper foil, 0.01 mm. thick, and the other of glass, through which the suspension is observed. The flakes orient themselves with their flat surfaces parallel to the wave front; but where the beam of vibrations (which may be focused by an "acoustic lens" of copper foil containing xylene) is interrupted by a flow the flakes remain irregularly distributed and the defect shows as a dark shadow in a bright field.

Pohlman worked with a 400-watt piezo-electric generator producing waves of frequency 800 kHz. In recent work D. S. Shrayber[5] has employed waves of 10,000 kHz frequency for the examination of metal parts with smooth surfaces and for the purpose of detecting small defects. While the directional property of the beam and the sensitiveness of the method are improved by increasing the frequency, the electrical equipment is more complicated and the scattering of the ultrasonic vibrations by the unevenness of the surface is increased. In this connection reference should be made to a valuable article which has recently been published by A. Behr[6] on "The Examination of Metals by Ultrasonics." This deals particularly with the instrumental side, the principles of construction of generators and detectors and the extent to which ultrasonic methods of non-destructive testing may be rendered available for the examination of metals for defects. It appears to be clear that the only efficient means of obtaining powerful mechanical oscillations of the desired frequency are the piezo-electric method and the magnetostriction method, whilst for the detection of ultrasonic waves of very short wave length only the piezo-electric method is sufficiently sensitive. The field which can be examined at one time is very restricted in comparison with the area that can be radiographed, and the method must therefore necessarily be slow. On the other hand, with the ultrasonic method it should be possible to detect fine cracks, for example, "flakes," situated at right angles to the beam in a considerable thickness of metal. Probably the most useful application of ultrasonics would be in detecting laminations parallel to the surface of a plate or forging. Unfortunately most of the published information seems to have been derived from the examination of "artificial defects," for example, holes drilled in bars, or plates built up in layers. These, however, are quite sufficient to demonstrate the practicability of the method and it seems probable that applications of ultrasonics in several directions only await the development on a commercial scale of suitable generators and detectors.

REFERENCES.

[1] The Iron Age, May 15th, 1941, page 48.
[2] The Metallurgist, December, 1938, page 177.
[3] "Ultrasonics and their Scientific and Technical Applications," by L. Bergmann (English translation by H. Stafford Hatfield, G. Bell and Sons, Ltd., London, 1938); "Supersonic Phenomena," by W. T. Richards, Review of Modern Physics, January, 1939, page 36.
[4] Zeit. fur Physik, 1939, 113, 697.
[5] Zavodskaya Laboratoriya, 1939, page 816.
[6] Metallurgia, November, 1940, page 7.

CONCRETE SHIPS.—Concrete ships or barges have recently been proposed to replace U. S. tanker tonnage removed from regular trade routes during the past several months. Mr. Rolt Hammond, who gives a brief

history of concrete ships in the following article reprinted from The Engineer, July 18, 1941, believes the concrete ship has a great future before it. Others state "there seems little reason to suppose that the present revival of the reinforced-concrete barge will outlast the period of the war, or that this method of construction will ever be able to compete successfully with steel under normal conditions." *

Present events are showing us all too clearly that the vital Battle of the Atlantic will only be won by a supreme effort in every direction, not least in the rapid replacement of lost tonnage; in this latter respect there is no doubt that the concrete ship will play an important part.

During the last war remarkable strides were made in developing this type of vessel, instanced by the fact that a total of 150,000 deadweight tons of concrete shipping was completed in Norway, Great Britain, the United States, France, Italy and in other parts of the world. The first concrete ship to be laid down was the Norwegian *Namsenfjord,* launched in 1917, which had a deadweight cargo capacity of 200 tons. On her official trials she showed remarkable freedom from vibration and steady steering in a south-westerly gale against heavy seas; this vessel was followed by the further Norwegian examples of the *Stier* and *Patent,* each of 600 tons, and then by the *Askelad* and *Concrete* of 1000 tons.

On her official trials the M.S. *Stier* ran into such rough weather that other ships of similar tonnage refused to put to sea. The Norwegian Government Inspector on board reported that the pumps were not used and that the hull remained perfectly tight. She was eventually sold to Greece, trading in the Mediterranean and Red Sea under the name of *Aliakmon.* Similar weather conditions were encountered by the *Askelad* during her trials in November, 1917; in December she sailed from Christiana to Rouen, through heavy seas in the teeth of a westerly gale. Her captain bore witness to the fact that she was thoroughly seaworthy; she was later engaged in trading to Norwegian, Swedish, Dutch and British harbors. She also made some voyages to the Baltic, where ice proved troublesome.

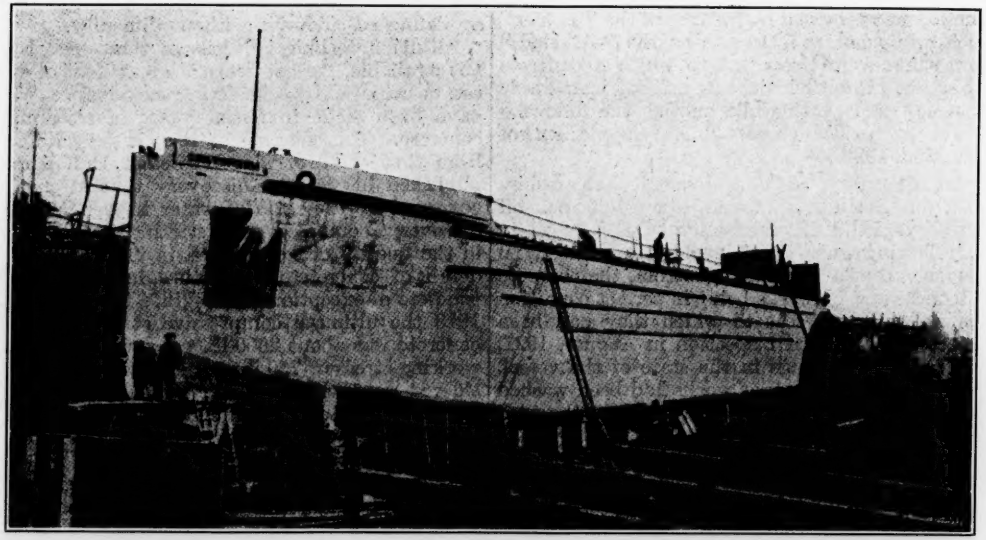
During the same period the British Ministry of Shipping embarked on a large program of construction, embodying about 150,000 deadweight tons; only about fifty seagoing lighters and twelve steam tugs were completed. Two outstanding examples of British-built ships provide evidence of behavior under severe conditions. These vessels were the steam tugs *Creteboom* and *Cretestem,* built in 1919 and examined in 1934, after more than fifteen years' service afloat.

The *Creteboom* sailed for some years in the Baltic, and on one occasion she was icebound for several days; after a special survey made by Lloyds on her return to this country certain restrictions on her trading movements were removed, and she afterwards traded in Portuguese waters. In 1926 she towed the Southern Railway Company's floating dock from the Tyne to Southampton.

Cretestem was also frozen up in the Baltic, and on one occasion, she came into head-on collision with a steamer. For several years these two ships were laid up in the Tyne during the shipping depression; although dirty and unkempt looking, their structures were perfectly sound. Expansion, due to the heat from the boilers, had formed hair cracks in the decks, none of which were of importance. Below decks neither weeping nor condensation were evident; the cabin walls, although dirty, only required a coat of paint. The bilges were also perfectly dry.

In the United States, where astonishing feats of rapid shipbuilding were accomplished by the United States Shipping Board, the concrete ship was developed to a surprising extent. Four cargo vessels of from 2500 to 3000

* Engineering (London), August 22, 1941, p. 150.



CONCRETE STEAM TUG "CRETEFARM" ON STOCKS.

STERN OF S.S. "ARMISTICE."

tons deadweight and eight oil tankers, each of 6500 tons, were completed before the Armistice. The most outstanding example was the *Faith*, a vessel of 5000 tons, launched at Redwood, California, in March, 1918. She made voyages to Vancouver, Honolulu and Chile, reaching New York harbour via Panama in November, 1918; in the spring of 1919 she arrived in London, the first concrete ship to cross the Atlantic.

Cost of repair is always a matter of vital importance, and here the concrete ship shows to advantage. For example, on January 16th, 1919, the motorship *Askelad*, during her passage through the English Channel under heavy weather, was driven ashore in the estuary of the River Somme; she bumped so violently on the bottom that footing could not be kept on the bridge. At low tide she was left high and dry and was abandoned by her crew, who believed that she would probably break up with the next tide. A few days later, however, it was found that the hull was quite sound, and she was refloated on the spring tide on February 1st. Ten days later she arrived in London, where a preliminary survey indicated that she was sufficiently seaworthy to go to Grimsby; she docked there on March 12th.

It is interesting to note that no repairs were needed on the concrete hull, beyond the application of a few patches of cement mortar by the crew. After a survey at Grimsby she was again put into service and was not dry-docked until the following September; minor repairs were done to deck fittings, and the cement finish on the deck was repaired in a few places. The total cost of repairing the hull of the *Askelad* after grounding on the French coast did not exceed £25; some years later the ship was engaged in the Baltic, North Sea and Channel trade.

Another interesting example of sturdy construction was provided by the case of the *Polias*, which, while not so fortunate as the *Askelad*, shows the severe treatment to which a concrete ship can be submitted without suffering irreparable damage. It is worth while quoting the following extract from "The Times Trade Supplement" of June 22nd, 1922:

"In February, 1920, a concrete steamship, 250 feet long, struck a submerged reef of rock off Port Clyde, U. S. A. Inspection of the vessel in the following July is said to have revealed no damage except a hole in the bottom, the concrete hull, decks and bulwarks being in perfect condition in spite of heavy seas to which they had been subjected. A second inspection in August, 1921, disclosed little damage in the state of the vessel. Some of the timber upper works had been washed away, but the hull appeared as solid as the rock on which it rested. According to recent reports the hull is still intact after exposure to the sea for over twenty-eight months. The chief of the salvage crew is reported to have stated that a steel or timber vessel would not have survived for more than three months in the same situation. The test is one showing very clearly the high resistance of reinforced concrete to exceptionally severe shocks and stresses."

Unfortunately, the veil of secrecy cannot be lifted to reveal the technical details of the work now being undertaken by the Admiralty in the construction of a large number of concrete barges having a carrying capacity of from 180 to 200 tons; these vessels have endured gruelling tests for seaworthiness and water-tightness with marked success. They are built up of pre-cast slabs made in a yard some distance from the shipways, the first operation consisting of laying the bottoms; these are formed of the slabs arranged in regular fashion and subsequently connected together by means of narrow concrete stiffening members cast in place. Slabs for walls and decks are joined in a similar manner, and in all cases the steel reinforcement projects beyond the edge of the slabs, so as to bond the slabs together like a monolith. For example, the sides of the barge consist of concrete slabs, which are connected at their lower edges to the bottom of the barge and at their vertical edges to one another.

Little skilled labor is required and the barges are launched sideways without difficulty.

Although technical details of these vessels are not available, we can learn a great deal of what can be achieved by considering the advances which have been made in recent years in research on concrete. The concrete of 1940 is very different from that of 1918, when Professor Duff Abrams published his famous discovery that strength is governed by the ratio of water to cement. This discovery, combined with striking improvements in the quality of cement and in practically every detail of concrete manufacture, has resulted in a concrete of exceptional strength. For example, in 1918 the ultimate compressive strength of good concrete was about 2000 pounds per square inch and a working stress of 750 pounds per square inch was considered somewhat daring; today, however, a compressive strength of 5000 pounds to 6000 pounds per square inch is expected in normal structural work, and greater strength is often obtained.

Recent bridges have been designed on the basis of a working stress of nearly 2000 pounds per square inch and 1200 pounds to 1500 pounds is common; great improvement has also been effected in steel reinforcement, a tensile stress of 27,000 pounds per square inch being commonly used instead of the 16,000 pounds used in designing the concrete ships of 1918.

Some indication of what may be achieved by modern methods is instanced by the recent developments of pre-cast concrete segments for lining the tube tunnels in London. Before segments of this type were manufactured on a large scale a great deal of research was carried out to determine the best method of procedure.

Employing an aggregate graded between ⅜ inch and ⅛ inch of washed shingle and washed sand, these are gauged by weight and the water content controlled to give a water/cement ratio of 0.42, "Ferrocrete" cement being used. A solenoid vibration table has been developed, the number of vibrations being 3000 per minute, giving an amplitude of 1/64 inch vertically and an acceleration of $4g$ to the particles. Segment and mould together are vibrated for a period of three minutes, the resulting density of the concrete being 150.4 pounds per cubic foot with a zero slump; compressive strength at seven days is 4000 pounds per square inch and 6000 pounds per square inch at twenty-eight days. The average compressive strength is 6500 pounds per square inch at seven days, a result which shows what can be achieved with vibrated concrete.

Future research in concrete will undoubtedly develop its possibilities even further, and the concrete of 1950 will be as far in advance of the present-day product as that material exceeds the 1918 concrete. Although great progress has been made in the production of light-weight aggregates, this lightness has only been achieved at the expense of strength, and so far no such material has been used for structural work. If these could be successfully developed, however, they would be of great benefit to the shipbuilder, who is very fortunate in being unhampered by many of the limitations which exist in building and constructional work. At present, light-weight construction must be looked for in improved quality of normal concrete and consequent reduction in thickness, rather than in the use of special aggregates.

PURIFICATION OF SALT WATER.—Although over 50,000 proposals have already been submitted to the Inventions Section of the British Ministry of Supply since the outbreak of war—6910 during the record month of June, 1940, following the evacuation of the British Expeditionary Force from the Low Countries, and the collapse of France—the following article, reprinted from the August 8, 1941 issue of Engineering reports a long-standing problem still unsolved.

There is, perhaps, no plight more pitiful than that of seamen adrift in mid-ocean with "Water, water, everywhere, Nor any drop to drink." The brutal German disregard of every humane tradition of the sea has, unfortunately, made such tragedies only too common. That men, surrounded by an unlimited expanse of water, should be condemned to perish miserably of thirst is a challenge to every engineer and physicist to find some means of preventing such happenings. The urgent need for some method of rendering sea water potable, which could be employed by the crew of a raft or open boat, was recently called to public attention by a correspondent of "The Times," and our contemporary emphasised the matter in a leading article. The publicity thus given to the question resulted in various suggestions being received, none of which unfortunately, seems to afford any practical solution of the problem. The provision of a small distilling plant, similar to those used on board ship, has obvious disadvantages. The weight and size of the apparatus, together with the necessity for carrying a supply of fuel for its operation, would appear to be prohibitive. It is obvious, moreover, that any distilling equipment that is carried will take up space that might otherwise be occupied by extra storage tanks, filled in advance with drinking water. This objection applies equally to all kinds of purifying apparatus, of which several alternatives to evaporators might be proposed. One correspondent suggested that, if sea water could be frozen, the ice, being free from salt, might be re-melted to form drinking water. Theoretically, such a procedure might be possible, but the provision of the power or the fuel to operate a refrigerating plant in a small boat would appear to be quite impracticable, apart from the bulk and weight of the equipment. Another writer thought that perhaps salt might be removed by some method of dialysis, but it would seem easier to get salt into water than out of it by this principle. A third made the pleasant suggestion that the salt could be precipitated by the addition of silver citrate to the water, converting the water into a sort of "synthetic lemonade" that would be innocuous and not unpalatable. It did not appear, however, that he had submitted it to a practical test.

Sea water varies in composition, but an average sample may be taken to contain about 2560 grains of dissolved solids per gallon, of which some 2000 grains are sodium chloride, *i.e.*, common table salt. The principal other salts in solution are about 240 grains of magnesium chloride, 150 grains of magnesium sulphate and 90 grains of calcium sulphate. Chemically pure drinking water is neither possible nor desirable, but how much of the above substances could be absorbed daily without harm to the human organism is a question for medical men rather than for engineers. The most heavily loaded public water supply of which we are aware in this country contains only 63 grains of dissolved solids per gallon, of which 18 are chloride of sodium, 18.5 are sulphate of magnesia and 20.5 are carbonate of lime. It is probably not necessary to attain anything like this degree of purity in an emergency supply of drinking water, and, indeed, many reputedly wholesome beverages are, in comparison, almost stiff with matter in solution; but 2000 grains of salt to the gallon, to say nothing of other chlorides and sulphates, appears to be definitely beyond the limit of what the ordinary human constitution can bear.

The field for devising means of reducing these quantities to a tolerable proportion is open to all who have a sufficient knowledge of the problem. Engineering, physical, and chemical schemes, as we have seen, have all been proposed, but the number of possibilities has by no means been exhausted. The ordinary salt, for example, might be removed by a base-exchange process, the converse of that which takes place in a zeolite water softener, but, unless there was some way of eliminating the resulting calcium chloride, the last state of the water would be as bad as the first. Alternatively, it seems conceivable that air could be saturated with the vapor of sea-water, and the moisture afterwards condensed out without too great difficulty. The

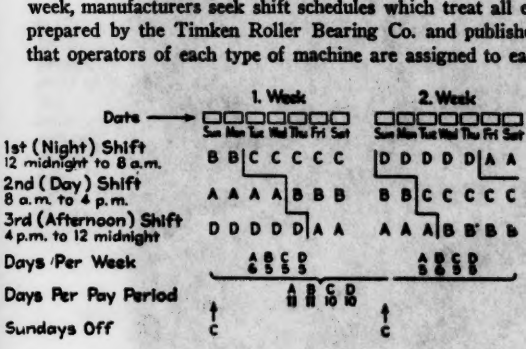
process would be a slow one, but experience has shown that even a few teaspoonsful of fresh water a day are sufficient to sustain life. The idea, at any rate, may be worth considering. There is, again the possibility that the salt content of sea water might be sufficiently reduced by some simple apparatus on the lines of the Hele-Shaw stream-line filter. We are under the impression, indeed, that some such claim was made for this device when it was first introduced, and it is certainly capable of filtering out every trace of color from water deeply stained by a fast dye, which is beyond the power of any ordinary method of filtration.

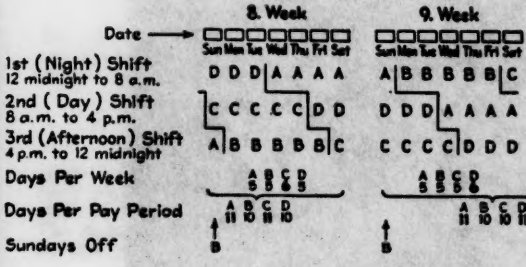
More hopeful than any of these, perhaps, is the process which was investigated some six or seven years ago in the Chemical Research Laboratory of the Department of Scientific and Industrial Research. It was demonstrated at the conversazione of the Royal Society in 1935 and briefly mentioned in the account of the exhibits which appeared in our issue of May 17 in that year. The process was based on the properties of certain synthetic resins of absorbing the various constituent salts in sea water from solution. Four tubes were employed, the first and third being filled with particles of a synthetic resin prepared from tannin materials, while the second and fourth tubes contained a synthetic resin prepared from aniline. Ordinary sea water was passed through the four tubes in series, the metallic constituents of the salts being removed by the first and third tubes and the acidic constituents, combined with the metals, in the other two. The water issuing from the fourth tube, we recorded, was drinkable and free from inorganic matter. After a time, the resins ceased to be effective, but they could be regenerated in a similar manner to that employed with the natural and artificial zeolites used in some water-softening plants. We were given to understand at that time that the process was still in the laboratory stage and that its economics had not been studied, and do not recall any report of later developments; but the cost should not be excessive, as the materials from which the resins were prepared cost only a few pence a pound, and the apparatus itself is of a simple character. Inherently, the problem of rendering salt water potable should not be more difficult of solution than that of revivifying exhaled and otherwise foul air, which may now be regarded as completely solved; and a little determined investigation along the various lines that seem most promising might well result in almost incalculable benefit to the seafarers to whom the nation owes so much at the present time.

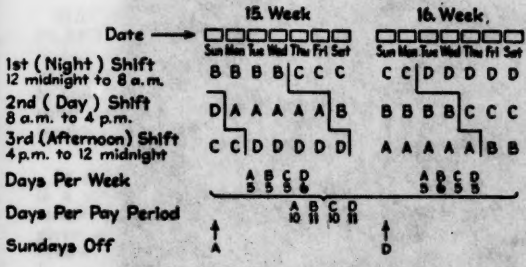
POSSIBLE DANGER WITH BLANK FLANGES.—This abstract of an article by E. Ingham, originally published in Boiler House Review in March, 1941, is here reprinted from the May, 1941, Transactions of the Institute of Marine Engineers.

Blank flanges are much used to seal the ends of steam pipes. Often, the sealing is done as a temporary measure whilst certain alterations to the plant are carried out, and there is a possible danger here in the fact that any convenient piece of flat plate may be used for making the flange, without due consideration being given to the important question of whether the plate is quite strong enough to resist the pressure. It should be realized that in the case of a large pipe under considerable pressure, the force tending to burst the flange may be several tons. Thus, a pipe 12 inches diameter has a sectional area of 113 square inches, and if the steam pressure is 200 pounds per square inch, the total pressure acting on the flange will be 22,600 pounds, or approximately 10 tons. It should also be realized that the flat form of plate is the weakest of all forms for resisting pressure, the hemispherical being the strongest. Hence, there may be real danger of explosion if a flat plate

CONVENIENT SHIFT SCHEDULE.—With m
week, manufacturers seek shift schedules which treat all e
prepared by the Timken Roller Bearing Co. and publish
that operators of each type of machine are assigned to ea

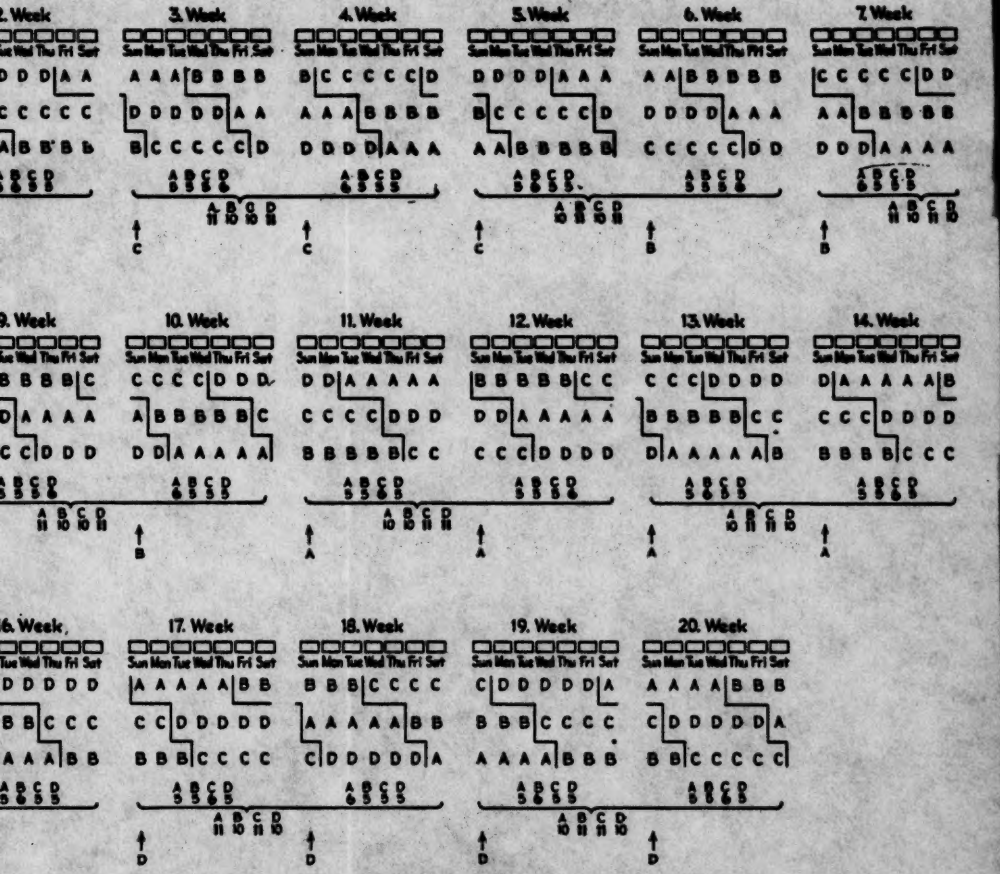






Crew A	Crew B	Crew C	Crew
1	1	1	1
2	2	2	2
3	3	3	3
4	4	4	4
5	5	5	5
6	6	6	6

E.—With many new factories engaged in defense work now operating twenty-four hours a day, seven days a h treat all employees equitably. This chart for a 4-crew set-up, with any number of men per crew, was and published by "Power" in the July, 1941, issue. In assigning men to crews, it is necessary to ascertain signed to each crew.



Crew D
1
2
3
4
5
6

NOTE THAT the three *shifts* are listed in the extreme left hand column; and that the four *crews* are designated by the letters *A*, *B*, *C*, and *D*. The hours designated by the letter *A* on the chart show exactly the hours and shifts crew *A* will work during the entire 20-week period, etc. Every man works five days in a row and then is off at least 48 hours, after which he changes shift. Over each period of 20 weeks every man has 5 Sundays off; over the same period every man works five extra days, totalling 105 shifts in 20 weeks. Over each period of 20 weeks every man has five pay days with 10 days pay and five pay days with 11 days pay, if he works full time. No relief men are required. Shift runs for 20 weeks then repeats without change.

is chosen at random to serve as a flange for sealing the end of a large pipe. In an actual case, where a 10-inch pipe carrying superheated steam at a pressure of 250 pounds per square inch had to be blanked whilst additional plant was being installed, a piece of ¼-inch hard brass plate was taken from the stores, and made into the required flange. The flange, which should have been approximately twice the thickness it was, failed, and as the consequence, three men lost their lives, and one was injured. In their book entitled "Steam Boiler Construction," the National Boiler & General Insurance Co. Ltd. give the following rule for determining the working pressure which may be safely imposed on the flat ends of cylindrical vessels up to 20-inch diameter and 1 inch thick, for any given thickness within the limit stated:

$$P = \frac{C \times t^2}{A}$$

where P = working pressure, in pounds per square inch,

C = a constant, 300,

t = thickness of plate, in sixteenths of an inch,

A = area of flat plate subject to pressure, in square inches.

The rule applies to mild steel having a tensile strength of from 24 to 28 tons per square inch. When the pressure is known, the required thickness of plate can be determined by the formula,

$$t = \sqrt{\frac{P \times A}{C}}$$

which is easily derived from the original. This useful rule is given because many engineers have difficulty in dealing with the strength of flat plates under pressure, the subject being complex if dealt with from first principles. The flange bolts, as well as the flange itself, must, of course, be of ample strength.

MACHINERY FOR THE POST-WAR GERMAN MERCANTILE FLEET.—In the following portions of a translated article, copied from the British Motor Ship for June, 1941, the author states his ideas of the types of machinery to be installed in the new German mercantile fleet which will be built up after the war concludes. Direct drive for motor ships is entirely ruled out although many owners of other nationalities have had sufficient confidence in the system to adopt it on a large scale and in important ships.

In the German shipbuilding journal "Schiffbau" for February 1, 1941, is a very long and detailed article entitled "Schiffsantriebsanlagen kommender Handelsschiffe" by H. Kedenburg, in which a careful analysis is given of the types of machinery that should be installed in the merchant vessels constructed after the war to rebuild the German mercantile marine. It is to be presumed that the article, which is given great prominence, represents the views of the leading German shipbuilders, engineers and shipowners. It is evidently based on the recognition that comparatively little of the pre-war German mercantile marine will exist, and that there will be need to take action without delay. A translation of parts of the article is given:

Immediately after the conclusion of the war, it will be the problem of the shipyards and the allied industries to supply groups of new ships to German owners as quickly as possible. The adoption of propelling machinery

of entirely new types or systems for the new merchant fleet is out of the question. The main question before the shipyards, owners and engine-builders is, therefore, to produce, by a combination of known machinery, greatly improved plant and an improvement of the general operation of propelling engines and auxiliaries. It is thus less a matter of the employment of entirely new devices than of the more effective utilization and combination of equipment and machinery with which experience has already been gained.

After the development of the two-stroke single-acting and two-stroke double-acting oil engine to a stage of complete reliability, a further step is the construction of the fast-running Diesel machinery for large and fast freight and passenger ships, with the object of saving valuable space. In order to obtain high propeller efficiency, some form of gearing is utilized between the engines and propeller shafts, or the Diesel-electric system is used.

The Machinery of Future Motor Ships.

The machinery installation of motor ships with relatively high power will, in the future, probably have the following characteristics.

1. Single-acting or double-acting two-stroke Diesel engines with Vulcan hydraulic drive or electric transmission to the propeller shaft.
2. Single-acting four-stroke Diesel engines for driving the auxiliary generators. The auxiliary electric machinery will derive its current from main engine driven generators when at sea.
3. The employment of exhaust gases for the generation of steam or hot water for heating purposes.

For given revolutions and power, the fast-running double-acting two-stroke engine occupies less space than the single-acting type. The height of the single-acting engine, particularly when a trunk piston machine, is, however, less, but the double-acting unit is shorter. The Diesel-electric drive or Vulcan hydraulic couplings will entirely displace the direct drive to the propeller, since the former systems have the advantage that the transmission of engine vibrations to the propeller shaft is prevented. The natural frequencies of the rotating parts of the propelling machinery are so high that the building up of dangerous resonance is not to be feared. Moreover, the periodicity of the power pulsations which the fast-running oil engine impresses on the hull sides are always higher than the natural periodicity of the hull itself. It may be accepted, therefore, that hull vibration does not come into consideration.

Through the adoption of the Vulcan gear, or electric transmission, it is possible to build motor ships with a very high propelling power. A multi-engine installation can be arranged with one, two or four shafts without any danger of torsional vibration. Further advantages of the hydraulic or electric drive over the direct coupled propelling installation are the following: The propeller shafting is not unfavorably stressed; in the event of the breakdown of one engine, this can be placed out of action and repaired, whilst the remaining machinery operates unhindered; overhaul of the main machinery can be carried out more conveniently, both at sea and in harbor; in freight ships with high machinery power the shaft tunnel is eliminated in the electric drive; the machinery can be tested without turning the propellers; the large number of small cylinders allows for the provision of lighter and cheaper spare parts, easier installation and replacement, smaller danger of cylinder and liner cracks arising through heat stresses, and the possibility of utilizing smaller and cheaper engine types.

Requirements for Steamers.

The propelling machinery of a steamer with moderately high power—about 5000 h.p. and upwards—should comprise the following elements:

High-pressure boilers with coal firing and mechanical stokers, or with oil firing.

High-pressure turbines or turbo-electric drive.

Electric drive for auxiliary machinery.

Generation of electric energy through a turbo-dynamo or Diesel generator.

Further use of "bled" steam for heating.

For large steam-driven passenger liners, the same conditions will apply, except that the "bled" steam should also be utilized for feed water heating and for other purposes on board. Oil-fired boilers alone would come into consideration, coal firing being ruled out.

Improvement in Silencing and Ventilation.

More attention will have to be given to the question of damping noise in the engine-rooms of the future merchant ships. The casings of gearing and other noise-generating equipment must be insulated, as it is necessary that the health and energy of the engine-room staff should not be detrimentally affected by excessive noise. This condition arises if it is impossible to be heard in the engine-room when speaking in natural tones, and the harm is increased if the degree of humidity and temperature in the engine-room are too high. Such conditions strengthen seagoing engineers in their desire to take up work on land. Particular attention should, therefore, be given to the question of engine-room ventilation at sea, and the possibility of the provision of air-conditioning plant for the crews' cabins should be investigated. In any event dining saloons and smoking-rooms should be air-conditioned.

Owners, engine builders and shipyards must, in the construction of the new merchant marine, give more consideration to the question whether it is not possible to develop further along standardized lines. Is it not feasible, for instance, when certain powers are in question, for the propellers to have standard hubs, in a number of ships? Moreover, propeller blades may be standardized with similar vessels, and by such methods the troubles when breakdowns occur will be diminished. Similarly, for main and propelling engines, standardization should be carried out to the highest possible degree, so that the greatest number of parts are interchangeable and the number of spares is reduced.

LUMINESCENT SIGNS.—The manifold uses of luminescent materials during war or peace need not be enumerated. This note is reprinted from Engineering (London), July 25, 1941.

The useful range of employment of luminescent materials, generally familiar in connection with watch dials, has been greatly extended by the conditions of nightly black-out. The after-glow properties of these materials may find valuable application for A. R. P. signs and shelter entrances, also for instrument boards with which, owing to difficulties of obscuration, ordinary shaded lights cannot be employed. By using phosphorescent materials, the field of supplementary low-power lighting may be considerably extended. Emergency illumination of this class may be used to furnish sufficient light in underground shelters to permit movement in case of failure of the ordinary lighting. Similar application to signs in factories will facilitate the evacuation of workpeople in case of damage to supply mains. Further possibilities are in connection with articles of wear for pedestrians and traffic-control signs.

The whole of this subject is dealt with in useful detail in Bulletin C. 17 of the Research and Experiments Department of the Ministry of Home Security. It defines luminescent materials as those which can be made to

emit light by means other than heating to incandescence. Excitation may be caused by X-rays, ultra-violet and visible radiation, by chemical changes and by friction, but only ultra-violet and visible-ray excitation is dealt with. The materials fall into two classes, fluorescent and phosphorescent. Fluorescent substances have the property of absorbing radiation within a certain range of wavelength, by this means being excited to emit radiation of longer wavelength. The fluorescence continues only so long as the material is irradiated. Phosphorescent material, on the other hand, continues to glow for some time after excitation has ceased. The duration of after-glow varies widely in different materials; in all, however, its intensity fades rapidly at first and then more slowly. British Standard Specification BS/ARP 18 distinguishes between two types of phosphorescent materials: those which have a high brightness during and for a short time after irradiation, but a short after-glow, and those which have at first a lower brightness but a longer after-glow.

Many organic materials, including various oils, greases and resins, are fluorescent. Some of them have been employed in the production of fluorescent paint, but there is a tendency for them to be washed out of the surface layer on weathering. To obtain prosphorescence, it is necessary to use inorganic compounds. Those most commonly employed are zinc and zinc-cadmium sulphides, and alkaline earth sulphides. The former belong to the first class of BS/ARP 18. With proper selection of materials, good durability can be obtained. Shielding from direct sunlight is generally desirable, otherwise darkening may occur, although it will not necessarily be permanent. Luminescent signs having a long life may be prepared by incorporating zinc sulphide in a vitreous enamel before firing.

Alkaline-earth sulphides belong to the second class, exhibiting lower initial brightness but longer after-glow. Calcium sulphides have good stability, but the commercially available material gives a deep violet glow to which the eye is relatively insensitive. Strontium sulphide gives an easily visible blue-green glow and has been largely used by manufacturers of luminescent paints. It is, however, unstable in presence of atmospheric moisture and careful precautions in manufacture, storage and use have to be taken. Special undercoats and protective varnishes are provided by manufacturers, but the Bulletin states that the most satisfactory system of protection is the enclosure of the strontium-sulphide paint between two sheets of glass with the edges sandwich waterproofed.

In the application of phosphorescent materials, irradiation is by daylight or artificial light sources, but for fluorescent substances continuous energizing by ultra-violet radiation is necessary. Mercury-vapor lamps with black outer bulbs may be used for this purpose. They are powerful sources of long-range ultra-violet radiation and a single lamp may be used to energize fluorescent markings over a wide area and at distances of 50 to 100 yards. Such lamps have a discomforting dazzling effect caused by the fluorescence of the eyeball of the observer and must be screened from direct view, and also from above. The argon glow lamp is another, but much weaker, source of ultra-violet radiation. It is convenient for energizing fluorescent signs at distances of a few feet.

Limits of brightness for standard A. R. P. signs are laid down in Specification BS/ARP 32. These limits are readily obtainable with a 5-watt argon lamp, but it is advisable to use an aluminium reflector to concentrate the ultra-violet radiation on the sign face. An objection is the restricted life of the argon lamp. A tungsten-filament lamp is superior in this respect, but the running expenses are greater, as a 40-watt tungsten lamp is necessary to give the same effect as a 5-watt argon lamp. For switch rooms in which ordinary lighting may not be practicable owing to difficulties of obscuration, the Bulletin suggests a mercury lamp and the marking of instruments and switches with a fluorescent material. If this is also phosphorescent it will

have the advantage of exhibiting after-glow should the source of irradiation temporarily fail. Zinc-sulphide pigments are recommended.

In connection with the use of phosphorescent substances for signs and obstruction marking, the Bulletin states that their limitations should be fully realized. None of the materials so far available can be relied upon to remain usefully visible throughout the night after excitation by daylight. With good quality strontium or strontium-sulphide, the markings will be usefully visible for about four hours after sunset on a moonless star-lit night. For several hours more they may be discernibly brighter than similar markings in white paint, but after that the white paint is equally, if not more, efficacious.

BOOK REVIEW.

LANGE'S HANDBOOK OF CHEMISTRY. FOURTH EDITION. PUBLISHED SEPTEMBER, 1941, BY HANDBOOK PUBLISHERS, INC., SANDUSKY, OHIO. 1874 PAGES.

Lange's Handbook of Chemistry, first published in 1934, has been revised and enlarged for the third time. The compiler, Norbert Adolph Lange, Ph.D., lecturer in Organic Chemistry at Cleveland College of Western Reserve University, was assisted as before by Gordon M. Forker, B.S., Chemical Engineer of the Harshaw Chemical Company. In addition to 1603 pages of tables of chemical data, the book contains an appendix of 271 pages of mathematical tables and formulas prepared by Richard Stevens Burlington, Ph.D., Associate Professor of Mathematics at Case School of Applied Science.

The Fourth Edition retains the conventional form of other handbooks which with its complex index and logical arrangement makes it extremely valuable to engineers who either frequently or occasionally need have recourse to chemical data.

The revision consists of the addition of eleven tables and the extension or complete rewriting of 15 tables. The volume continues to keep pace with the development of Chemical Engineering.

ASSOCIATION NOTES

ANNUAL MEETING.

The annual business meeting of the Society was held in Washington, D. C., on Tuesday, October 7, 1941, pursuant to its By-Laws. The meeting was unusually well attended.

NOMINATIONS FOR OFFICERS FOR 1942.

The following have been nominated for offices of the Society for the calendar year 1942:

For President:

Rear Admiral Herbert S. Howard, U. S. Navy.

For Secretary-Treasurer:

Commander J. E. Hamilton, U. S. Navy.
Lieutenant Commander Edward C. Forsyth, U. S. Navy.

For Member of Council:

Captain Bryson Bruce, U. S. Navy.
Captain S. M. Kraus, U. S. Navy.
Captain H. E. Saunders, U. S. Navy.
Commander L. F. Small, U. S. Navy.
Commander E. W. Mills, U. S. Navy.
Lieut. Commander S. N. Pyne, U. S. Navy.
Commander W. J. Kossler, U. S. Coast Guard.
Lieut. Commander R. B. Lank, U. S. Coast Guard.
Mr. Roger Williams.
Mr. Austin Kuhns.
Mr. P. N. Israel.
Mr. James Bates.
Mr. Rudolph Michel.
Mr. R. N. Landreth.

Ballots have been distributed. Polls close at 3:30 p.m., December 26, 1941.

ANNUAL BANQUET.

The annual banquet of the Society will be held at the Willard Hotel in Washington, D. C., on Saturday, May 9, 1942. Details will be announced later.

LIFE MEMBERSHIP.

A proposed change in the By-Laws of the Society establishes a Life Membership, at a cost of $150.00. Provision for voting on this question has been made on the ballot for officers for 1942.

MEMBERSHIP.

The following have joined the Society since the publication of the August, 1941, JOURNAL:

NAVAL.

Barnhart, Robert E., Ensign, U. S. N. R., Ship Superintendent Office, Navy Yard, Pearl Harbor, T. H.

Boone, Kenneth I., Lieutenant, U. S. N. R., U. S. S. *Williamsburg.*

Buerschinger, Wallace R., Ensign, U. S. N. R., 10 Pleasant St., Stamford, Conn.

Ciccolella, J. A., Lieutenant, U. S. C. G.

Clark, S. F., Lieutenant, U. S. C. G.

Colmar, Peter V., Lieutenant, U. S. C. G.

Craig, John R., Lieut. Commander, U. S. N. R., 20 Bedford Court, Wilmington, Del.

Dodge, Jesse K., Lieutenant, U. S. N. R., U. S. S. *Dobbin.*

Gill, Irving L., Commander, U. S. C. G.

Hankinson, R. L., Commander, U. S. C. G.

Harding, L. M., Lieut. Commander, U. S. C. G.

Hartung, Walter P., Lieutenant, U. S. N. R., U. S. S. *Mackinac.*

Hoag, Edgar L., Lieutenant, U. S. N. R., Cramp Shipbuilding Co., Richmond and Norris Sts., Philadelphia, Pa.

Hughes, Robert, Lieutenant, U. S. N. R., Office of Superintendent of Shipbuilding, Dravo Corp., Neville Island, Pittsburgh, Pa.

Kelley, John E., Lieutenant, U. S. N. R., 3809 31st St., West, Seattle, Wash.

Labberton, J. M., Lieut. Commander, U. S. N. R., Associate Professor, M. E., New York University, University Heights, New York, N. Y.

Pollak, Edward George, Ensign, U. S. N. R., 16 Canacao Boulevard, Navy Yard, Cavite, P. I.

Prescott, Elmer L., Chf. Mach., U. S. N.

Roberts, R. J., Lieutenant, U. S. C. G.

Rossheim, D. B., Grad. N. A., Consulting Engineer on Pressure Equipment and Materials, The M. W. Kellogg Co., 225 Broadway, New York, N. Y.

Sargent, Thomas Reece, Lieutenant, U. S. C. G.

Sexton, Frank M. P., Ensign, U. S. N. R., Ship Superintendent Office, Navy Yard, Pearl Harbor, T. H.

Sims, William E., Ensign, U. S. N. R., U. S. S. *Crosby.*

Slaven, Franklin W., Lieut. Commander, U. S. N. R., U. S. S. *Tuscaloosa.*

Vincent, Robert E., Ensign, U. S. N. R., U. S. S. *Spica.*

Walker, A. W., Jr., Lieutenant, U. S. N.

Walton, Fred W., Lieut. Commander, U. S. N.

Wells, J. Arthur, Ensign, U. S. N. R., U. S. S. *Barnegat*

Willard, J. B., Ensign, U. S. N. R., Bureau of Ships, Navy Dept., Washington, D. C.

Williams, G. Bullitt, Ensign, U. S. N. R., 710 Ward St. Greenville, N. C.

Yates, Russell E., Lieutenant, U. S. C. G.

CIVIL.

Harkrader, John P., Sales and Engineering Installation, Welin Davit and Boat Corporation, P. O. Box 28, Perth Amboy, N. J.

Huet, George O., Naval Architect, Higgins Industries, Inc., New Orleans, La.

Kennedy, William M., President and General Manager, Condenser Service and Engineering Co., 335 Lyndhurst Ave., Lyndhurst, N. J.

Nielson, M., Jr., Supt. of Marine Erection Dept., The Babcock & Wilcox Co., Room 1001, 85 Liberty St., New York, N. Y.

Sherman, Arthur L., Sales Engineering, J. S. Coffin Jr., Co., 326 South Dean St., Englewood, N. J.

Taylor, W. F., Sales Engineer for N. E. States, Allis Chalmers Mfg. Co., Room 1160, Park Square Building, Boston, Mass.

Werkenthin, Theodore A., Materials Engineer, Bureau of Ships, Navy Dept., Washington, D. C.

ASSOCIATE.

Dake, Lawrence, Engineer, Cargocaire Engineering Corp., 15 Park Row, New York, N. Y. Mail 405 W. 23rd St., New York, N. Y.

Dych, Harry L., Quarterman Machinist, Navy Yard, Philadelphia, Pa., Mail 5444 Rutland St., Philadelphia, Pa.

Karr, Milton, Engineer, Consolidated Steel Corporation, Shipbuilding Division, Orange, Tex. Mail 804 Orange St., Orange, Tex.

Mailander, John Hugh, Supt. Pipe and Fob. Div. Noland Co., Inc., Newport News, Va. Mail 725 N. Abington St., Arlington, Va.

Rettaliata, J. T., Asst. Engineer, Steam Turbine Dept., Allis Chalmers Mfg. Co., Milwaukee, Wis.

Smith, John S., 937 Spruce St., Philadelphia, Pa.

Taylor, Walter F., Hydraulic Laboratory, Newport News Shipbuilding & Dry Dock Co., Newport News, Va.

TRANSFERRED ASSOCIATE TO CIVIL.

Bushby Richard F., Chief Quarterman Machinist in Charge Engineering Installation, Navy Yard, Philadelphia, Pa., Mail, 1544 Minnesota Road, Camden, N. J.

Nicholson, Robert, Chief Quarterman Machinist, In Charge Ordnance Installation and Testing, Navy Yard, Philadelphia, Pa. Mail 232 Mountain St., Philadelphia, Pa.